International Marine Engineering

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THE SHIPBUILDING AND ENGINEERING COMPANY OF BURMEISTER & WAIN.

BY AXEL HOLM.

This company, which employs 3,000 workmen, made its first appearance in 1846. The firm then was called the Baumgarten & Burmeister Engineering Company, and employed only thirty men in a little workshop in the middle of the town (Copenhagen, Denmark). In 1851 the firm leased a small shipbuilding yard near the present company's engineering department, in the Copenhagen inner harbor, on the Christianshavn side, where they launched their first wooden steamship in 1854, the paddle steamer *Hermod,* and from this unpretentious beginning grew gradually the present large company. In 1872 the present yard on the island of Rephalen in the outer harbor was established, and at the same time the whole work was transferred to a joint stock company, under the name of the Burmeister & Wain Shipbuilding & Engineering Company. The first ships launched here were ships Nos. 83 and 84, the steamships *Christiania* and *Christian IX.,* on the 12th of April, 1874.

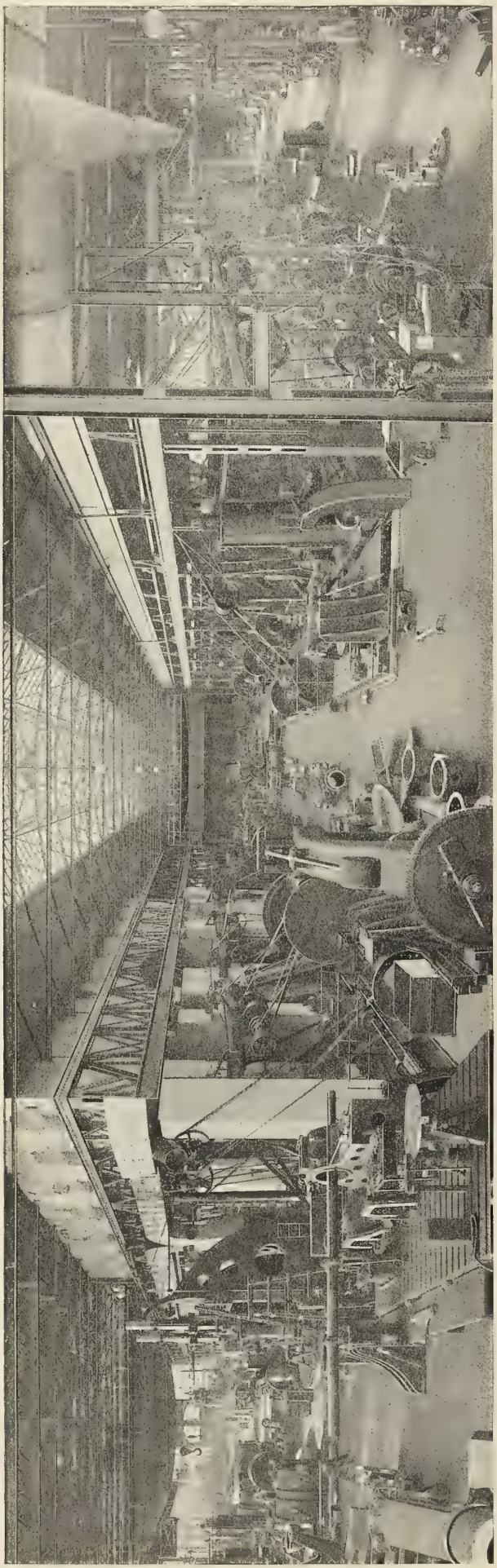
Now the company occupies in the inner harbor an area of 9 acres, where only the engineering departments are situated, and on the island Rephalen 18 acres of land, a partially filled up area, and 6½ acres water area belongs to the shipbuilding department.

As may be seen from the accompanying plan, the yard possesses four large building berths, three slips, a graving dock and a large floating dock. The building berths are 420 feet, 350 feet, 300 feet and 250 feet in length, with all necessary derrick cranes and electric winches for handling the materials;

THE FLOATING DOCK IN THE OUTER HARBOR AT COPENHAGEN.

the slips are two, 275 feet in length and one 310 feet, operated by hydraulic power. The graving dock, which is capable of holding ships of 10,000 tons, is 470 feet in length, 64 feet 9 inches in breadth and 23 feet depth at sill, and is divided into two smaller compartments by steel caissons. The dock was built in 1894, of granite concrete (in this year a great fire burst out on the island and partially damaged the gates under construction). Of the four steam centrifugal pumps for this dock are three with 20-inch suction and one with 8-inch suction, the common discharge channel being 3 feet 6 inches by 2 feet 3 inches. On the dock quays are traveling cranes of 15 tons capacity.

In 1906 a great floating dock was purchased in Antwerp



THREE VIEWS IN THE ENGINEERING SHOP OF BURMEISTER & WAIN, SHIPBUILDERS, COPENHAGEN, DENMARK.

ON THE HAULING-OUT SLIPS.

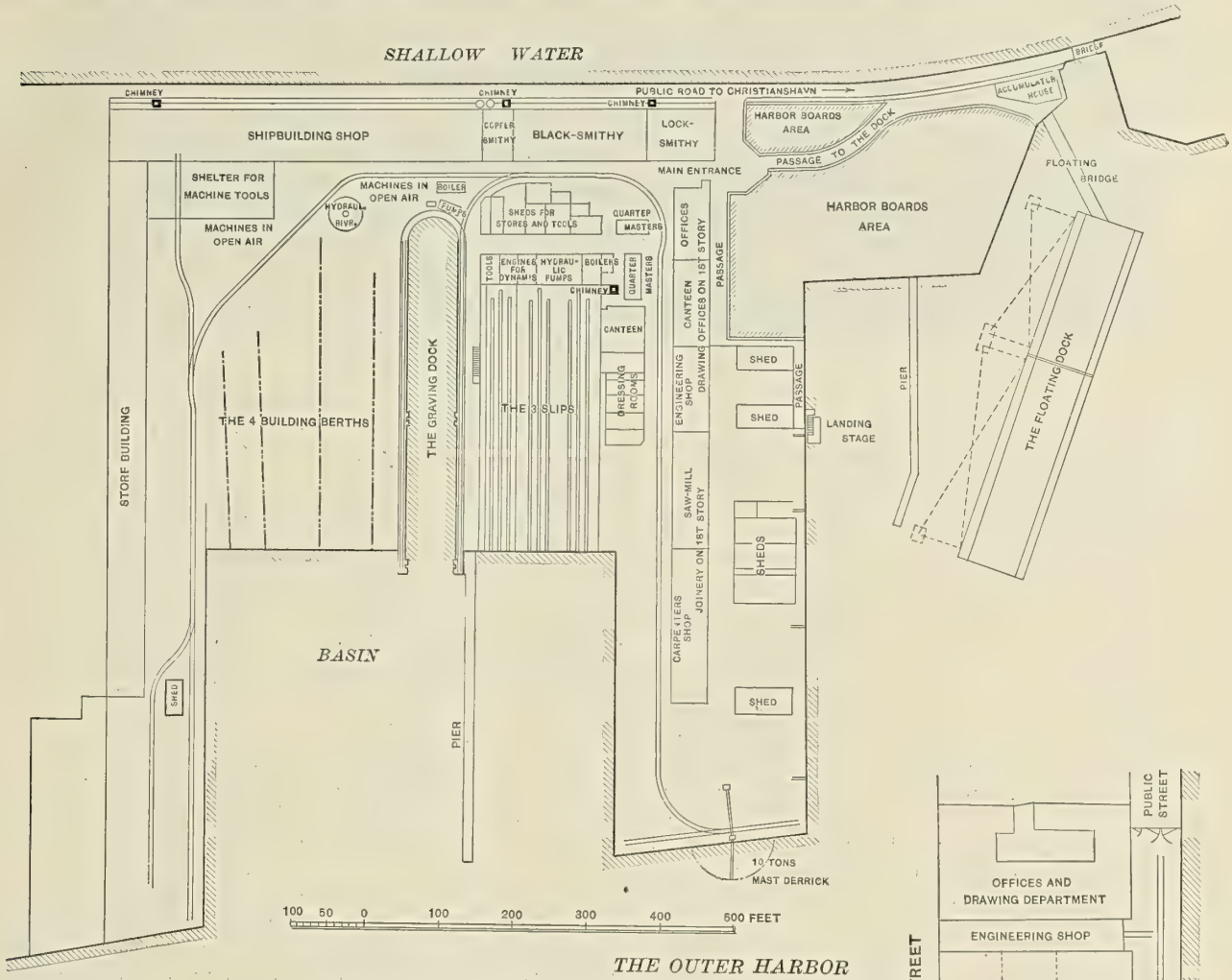
and towed to Copenhagen. This dock measures 493 feet in length, 98 feet in breadth outside, 70 feet inside and 23 feet depth over keel blocks. The dock's lifting capacity is 11,500 tons, and it is divided into two parts, 3/7 and 4/7, for two smaller vessels. The machinery is all newly installed and works by electricity. There are seven centrifugal pumps, operated by electric motors, at 220 volts and 440 amperes, and six winches for hauling and hoisting purposes. The lighting is also electric, and means are provided for the use of electric drills, as also for pneumatic tools; all power is developed from the yard's power station on shore. The passage from dock to yard is provided for by means of a 17-foot wide floating bridge.

In respect to tools, machines and cranes the yard's whole outfit is up to date. There are three steam cranes of 5 tons lifting power, traveling on a wide gear system of rails; one mast derrick, 100 feet in height, with a lifting capacity of 10 tons, and several small derricks and overhead cranes for hand and electric power. The power plant consists of two steam dynamos of 400 horsepower and one of 300 horsepower, as also a great accumulator battery and an air compressor of 150 horsepower. There are three Babcock & Wilcox water-tube boilers of 1,850 square feet heating surface and one of 90 square feet, together with a marine boiler of 90 square feet, all working with a pressure of 150 pounds per square inch. The machine tools are all driven by separate motors; 220 volts are used for power and 110 volts for lighting.

Several punching and shearing machines for plates and profiles, hydraulic manhole punchers, pressers and riveters, a hydraulic cold flanging machine, plate bending rolls and manglers, edge planing machines, a plate joggling machine and a corner planer for joggled plates, beam benders, beveling machines, saws, drills and countersinkers, etc., all of English make, are distributed in the shipbuilding shop at the east side

A CORNER OF THE SHIPBUILDING SHOP.

of the yard and in open air. In the shop also are installed the great plate and angle oven, the bending slabs and the scrive boards. In connection with the construction department are built small shops for coppersmiths, blacksmiths and locksmiths, these departments having their main shops at the engineering department on Christianshavn. At the yard's northern side is the large store building, and at the south we find another large building sheltering the joiner shops, the carpenters' shop, a small engine repairing shop, the canteen and the drawing offices, etc.



SHIPBUILDING DEPARTMENT ON THE REPHALEN, COPENHAGEN.

ENGINEERING DEPARTMENT OF BURMEISTER & WAIN, CHRISTIANSHAVN, COPENHAGEN.

The engineering department on Christianshavn is also up to date, owing to the several great newly-erected shops, large purchases of new pieces of land, newly-provided modern machine tools, enlargements, etc. The power distribution here

GENERAL VIEW OF THE SHIPYARD OF BURMEISTER & WAIN, COPENHAGEN.

is also electrical throughout, direct-current motors being used.

In the power station are installed two vertical triple expansion engines of about 400 horsepower, and one large horizontal compound tandem engine of about 2,000 horsepower (the greatest of this type in Scandinavia and from the company's own works), all coupled to direct-current dynamos, giving 220 volts for power and 110 volts for lighting. The boilers are: one Babcock & Wilcox of 435 square feet heating surface; two Lancashire boilers of 285 square feet each, and three other Babcock & Wilcox, each of 722 square feet heating surface; all with mechanical stokers and superheaters. The working pressure is 150 pounds. There is also installed an air compressor of about 150 horsepower. On rails running between all departments travel two steam "jumboes" of 3 and 7 tons capacity, and one locomotive for transport of boilers, scrap and heavy work.

The new and fully modern engine shop is divided into three bays; the main bay is 350 feet in length, 67 feet 6 inches in breadth and 42 feet 6 inches in height. One side bay is 260 feet in length, 51 feet in breadth and 40 feet in height; while the other side bay is 260 feet by 56 feet and 23 feet in height, and over this bay is installed the large cream separator works in two stories, each about 16 feet in height. In the engine shop are three electric overhead cranes of 30, 20 and 12 tons lifting power, with auxiliary lifts. The machine tools are all new and up to date, with separate electric motors and of English, German and American manufacture. Among others is installed a large turning lathe for propeller shafts, driven by a 7-horsepower motor—the largest lathe in Scandinavia. Another lathe is capable of turning shafts 65 feet in length, etc., all working with high-speed steels.

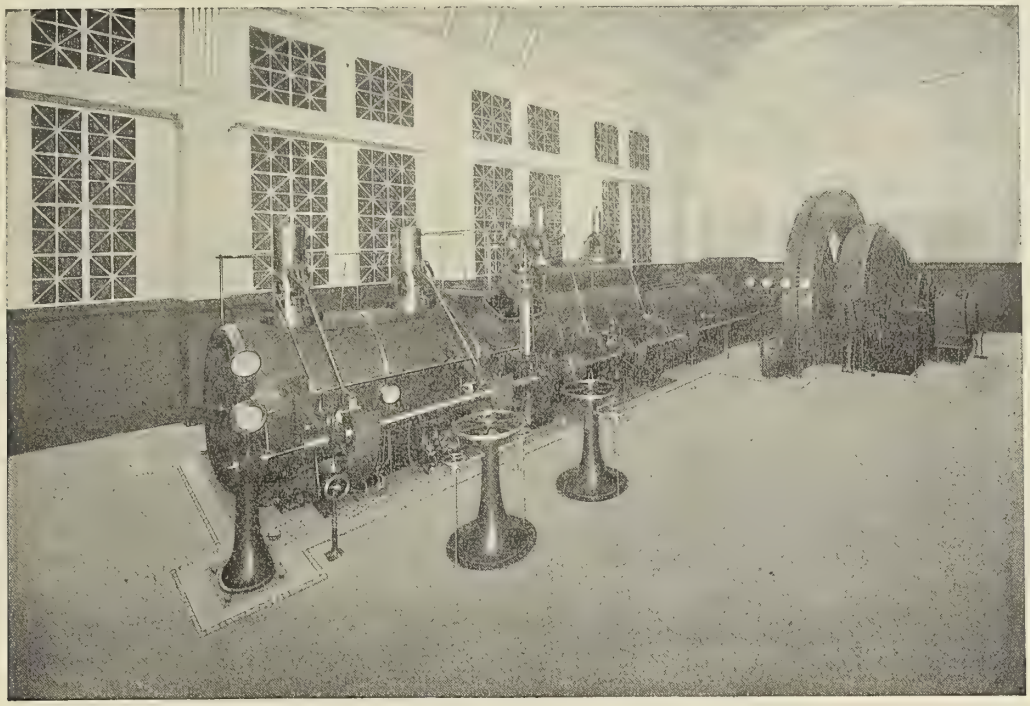
The shops are wooden paved throughout, and dressing rooms and lavatories are provided for the laborers' use. The heating is through hot air (the Buffalo system), 4,600,000 cubic feet of hot air being delivered per hour; the exhaust steam from presses and steam hammers is used for this purpose.

The boiler works occupy two shops, 295 feet in length, 49 feet 3 inches in breadth and 42 feet 9 inches in height, and here are arranged two overhead electric cranes of 10 and 40 tons lifting power, also with auxiliary lifts, besides two smaller ones for 3 and 5 tons. The shops are fitted with hydraulic riveting engines in pits, vertical plate rollers, cold flanging machines and several boring and planing machines, etc. Also, there is installed a large plate oven, heated by gas. The work is capable of delivering all kinds of boilers up to the largest sizes in use.

The great forge shop is 390 feet by 51 feet by 72 feet in height. The two overhead cranes here are of 35 and 20 tons capacity, the 35-ton crane having an auxiliary lift of 5 tons capacity, and several other small cranes are installed. There are seven steam hammers of different sizes and makes, with the accompanying furnaces. Three gas generators supply gas to six heating furnaces for large forge pieces, which are worked out by means of two hydraulic presses of 1,200 and 2,000 tons pressure, respectively. There is in addition another hydraulic press of 800 tons pressure, with separate coke furnaces, and a 6-ton crane. The presses are all of German make, and the 2,000-ton press has just been installed. The manufacture of shafts and heavy forge pieces has, through these excellent tools, increased to about 3,000 tons a year in 1906, against 1,000 in 1899. A great deal of the output is exported to Scotland, England, Norway, Finland and North Germany. The ingots for these forgings were formerly taken from Germany, but now a large Martin steel works is under erection.

The great foundry is an almost new building, erected in 1900, of steel framework, with brick filling (the engine shop also is partially steel and brick-built). The building is 280 feet in length, with three bays, the middle bay being 52 feet

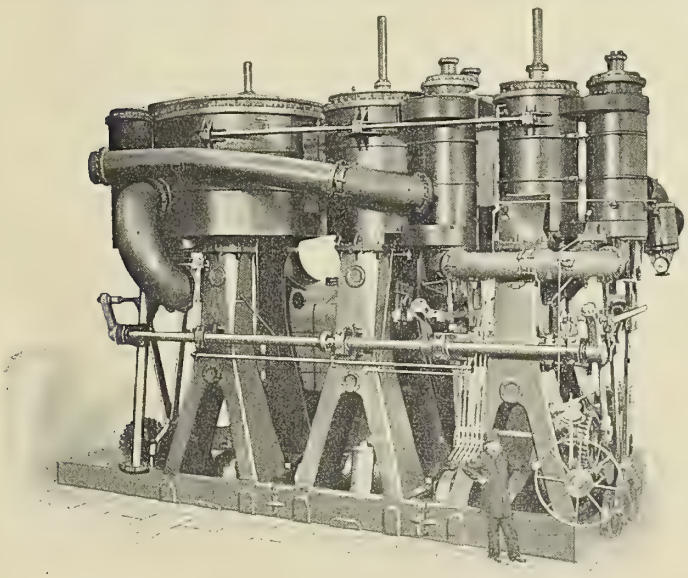
THE ENGINEERING DEPARTMENT AT CHRISTIANSHAVN, IN THE INNER HARBOR.



THE 2,000-HORSEPOWER ENGINE AND DYNAMO IN THE ENGINE ROOM.

6 inches broad and 46 feet high, and the two side bays 32 feet 9 inches broad by 26 feet 3 inches high. Here travel three overhead cranes of 20, 20 and 10 tons lifting capacity. Three cupola furnaces, two drying hearths and several casting pits are arranged for, and the foundry can deliver pieces up to 25 tons. A metal foundry is also located here. The sand blasting for cleaning takes place in a separate house close to the foundry.

In connection with this iron foundry is the new Siemens-Martin plant under erection. It is planned for a yearly output of 10,000 tons, and the shop, built of steel framework, has a ground area of 148 feet by 72 feet 6 inches, and is 51 feet high. The shop will be equipped with one Martin furnace of Swedish pattern for charges of 20 tons, with four generators producing the necessary gases. For handling the raw material, the scrap and pig iron, a 6-ton electric molding crane and electric elevators are to be installed, together with the necessary cranes for the coal and scrap from barges. In the shop will also be arranged an overhead electric crane of 35 tons lifting power, with an auxiliary lift. The work is intended only to deliver ingots for forgings and cast steel pieces.



MAIN ENGINE FOR IMPERIAL YACHT STANDART.

In the Christianshavn plant are a testing house for Diesel motors, a model joinery with roomy storehouses, stores for cream separators, a laboratory, a testing dairy, coppersmithy and painters' shop; and close to the main entrance are situated the large and well-lighted drawing offices (for about thirty men) and the several managing offices for both yard and engine department.

Among the several prominent works done by the firm may be mentioned the Russian imperial steam yacht *Standart,* the Russian cruiser *Bojarin* (lost in the Russian-Japanese war), together with several large cargo and passenger steamers, and the whole engineering outfit for the large pumping stations for the Copenhagen drinking water supply, and for the electric power and light centrals of Copenhagen, Stockholm, St. Petersburg and Baku, etc.

THE SHEARING STRENGTH OF RIVETED SEAMS IN SHELL PLATING.

BY R. E. ANDERSON.

In the design of vessels of certain types it frequently becomes desirable to investigate the longitudinal shearing forces which will obtain in the seams of the shell plating, in order to be certain that sufficient strength is provided in the riveted connection. The exact nature of these longitudinal or horizontal shearing forces is not always clearly appreciated. The consideration of one or two simple examples should, however, make the matter clear.

If we take a book having limp covers, or a pile of paper or cards, and bend it, it offers very little resistance, and we notice that the leaves or the cards slide slightly upon each other. This is because each leaf retains its original length; there is nothing to make the leaves on one side compress and those on the other side stretch, as they would if they were well glued together, and so each leaf bends separately.

Similarly, if we take four or five boards, and, laying them

one on top of another, use them thus as a beam to support a load, they will bend considerably if the load is a heavy one. If we have made some pencil marks on the edges of the boards we will find that there has been a slight sliding of the boards upon each other, and that this sliding is greatest near the ends. We can prevent the sliding by spiking the boards together. If we put in only one or two spikes we shall not prevent the sliding, because the wood around each spike will crush, or perhaps, if we have used wooden dowels instead of spikes, the dowels will break; but if we put in enough spikes or dowels we can prevent the sliding altogether, and the beam will not bend under the load nearly as much as it did before the spikes were put in. The tendency for the boards to slide upon each other will be just as great as before,

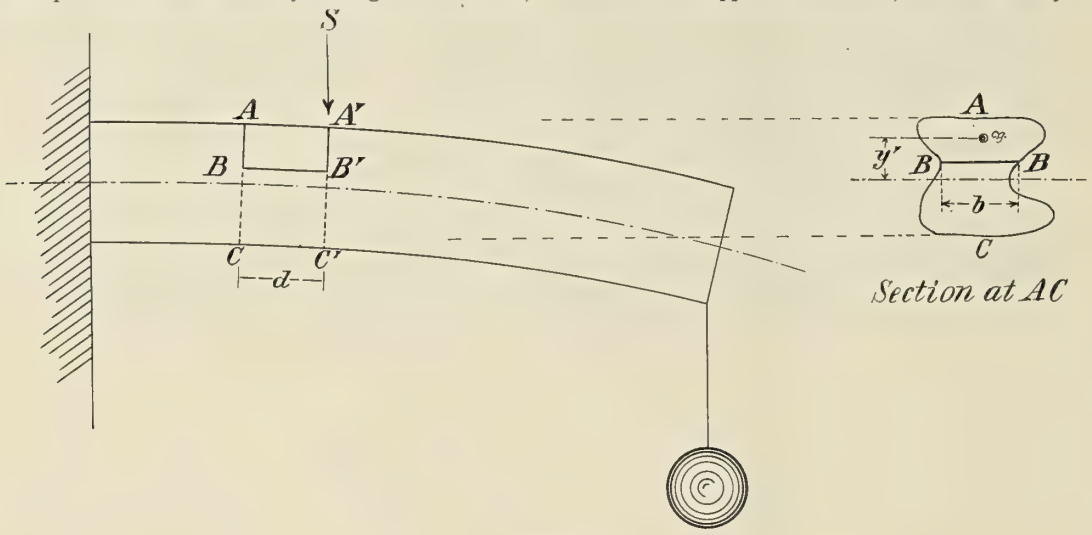


FIG. 1.

but it will be taken up by the spikes. This tendency to slide is the longitudinal shearing force, and it develops a shearing stress in the spikes. If the beam had been a solid piece of timber instead of being made up of a number of layers, the longitudinal shear would still have existed, but it would have been resisted by the shearing strength of the timber itself instead of by the spikes.

It will be seen that the capability of the spikes to resist the shearing force determines whether the beam will act as a unit, or only as several thin boards. Before we spiked the boards together the beam could bend by bending each board independently; after we put the spikes in the beam could bend only by stretching the boards on one side and compressing those on the other. It is evident, then, that if a complicated riveted structure is to act as a unit, and is to obey the laws upon which the ordinary formulæ for the strength of a girder are based, and which assume that the stresses and the consequent stretch and compression of the material are in direct proportion to the distance from the neutral axis, its various parts must be so connected together that their tendency to slide upon each other is efficiently resisted. In order to provide for resisting the tendency to slide, we must know how great that tendency is; in other words, we must know the value of the longitudinal shearing force.

DERIVATION OF FORMULA.

Consider, for example, the upper side of a beam which is so loaded as to put the lower side into compression and the upper side into tension; that is, it is fixed at one end and loaded at the other (Fig. 1). Assume that we wish to investigate the forces acting on the plane BB', which is somewhat above the neutral axis; and in order that our problem may not be unduly complicated we will let BB' be relatively very short.

Now, all the fibers above BB' are in tension, but since the bending moment at B is greater than at B', the tension on any given fiber is greater at B than at B'. Thus, on the plane AB there is a total pull which is greater than the pull on $A'B'$, and were there nothing to resist it the whole block $ABB'A'$ would begin to move to the left. It would not be prevented from moving by the portion of the beam to the left, nor by the portion to the right, for a precisely similar tendency exists for these portions of the beam to move also in the same direction. Motion is, however, prevented by the fact that the portion below BB' has much less tendency to move, or rather, considering the entire portion below BB' to the lower edge of the beam, has an exactly equal tendency to move in the opposite direction, and the ability of these portions to resist their tendency to slide upon each other constitutes the longitudinal shearing strength of that part of the beam on the plane BB'.

It is necessary here to assume acquaintance with the elementary formula for the stress in a beam:

$$p = \frac{M\,y}{I}$$

where p is the intensity of the stress in any fiber distant y from the neutral axis. M is the bending moment on the beam at the point in question, and I is the moment of inertia of the cross section of the beam.

Now to determine the *whole amount of pull* on the portion AB of the beam, we must multiply together the area of that portion by the average intensity of the pull. This average intensity will be the same as the intensity of the pull on that fiber which is at the center of gravity of the portion we are considering. We must therefore multiply together the number of square inches in the portion ABB and the pull per square inch at the center of gravity of ABB.

Now let y' be the distance from the center of gravity of ABB to the neutral axis, and let a be the area of ABB. Then the average pull per square inch is

$$p = \frac{M\,y'}{I},$$

and the whole pull on ABB is

$$P = \frac{M\,y'\,a}{I}.$$

Likewise, the whole pull on $A'B'B'$ is

$$P' = \frac{M'\,y'\,a}{I}.$$

The difference between these pulls, or the whole shearing force on the plane BB' (which force we will call F), is

$$F = \frac{M\,y'\,a}{I} - \frac{M'\,y'\,a}{I},$$

or

$$F = \frac{y'\,a}{I}\,(M - M') \ldots\ldots\ldots\ldots (1)$$

It is now necessary to find the value of $M - M'$. Consider again Fig. 1. We can cut the beam through at $A'C'$, and leave the conditions at AC unaffected, provided we replace all the internal forces acting on $A'C'$ by external forces of like nature and value. We have on $A'C'$ a direct pull above the neutral axis, varying in intensity from nothing at the neutral axis to a maximum at A', and this we can replace by

$$F = \frac{y'\,a}{I}\,Sd \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

This gives the whole shearing force on the plane BB'. To get the stress per square inch we must divide by the area affected, which is $b \times d$, where b is the breadth of the beam at the point in question.

We observe also that the quantity $y'\,a$ is the statical moment of that portion of the cross section of the beam which is outside of the plane of shear BB'. Representing $y'\,a$ then by m, and dividing by $b \times d$, we have finally for shearing stress per square inch on any longitudinal plane at a given distrance from the neutral axis:

$$f = \frac{mS}{Ib} \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

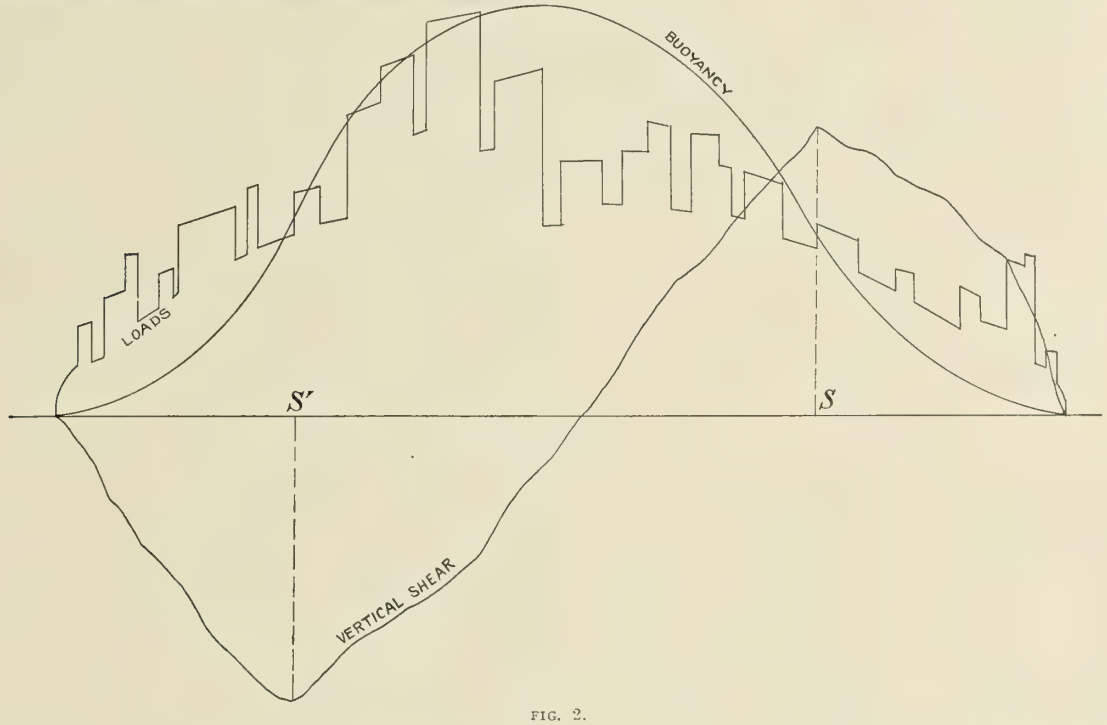


FIG. 2.

an external force or a series of forces acting in the direction of the beam. In like manner we can replace the compressive forces below the axis by external forces also acting in the direction of the beam. There remains, however, the vertical shearing force tending to make the portion to the right of $A'C'$ slide downwards. This may be replaced by an equal external force, which we will call S, the vertical shear on the section. These three forces, or series of forces, completely replace all those acting on the plane $A'C'$, and the conditions at AC will therefore be unchanged.

Now, the bending moment on AC must be the sum of the moments of these forces. The moments of the pulling forces above the axis, and of the compressive forces below the axis, must be the same at AC as at $A'C'$, since these forces act in the direction of the beam, and their sum is M', since it was M' that they replaced. If d be the distance between AC and $A'C'$, the moment of S is Sd.

We have, therefore, for the bending moment at AC,

$$M = M' + Sd,$$

whence

$$M - M' = Sd \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

This is true so long as d is a comparatively short distance, or rather so long as d is not too great for the force S to be regarded as practically constant. Putting this into equation (1) we have

which is the general equation for longitudinal shear in a beam, and may be stated thus:

To find the intensity of the longitudinal shear at any point in the cross section of a beam, multiply the total cross shear on the section by the statical moment of that portion of the section which lies outside the point in question, taking this moment about the neutral axis, and divide by the product of the breadth of the beam at the point in question and the moment of inertia of the whole section.

It will be noticed at once that the value of m is greatest for the neutral axis itself, and that, consequently, for any beam the longitudinal shear is greatest at the neutral axis, the exception being a beam which is locally very thin at some point not very far from the neutral axis, in which case the maximum shear may come at this point.

In the derivation of these equations the use of the calculus has been purposely avoided. Those who prefer the more rigid mathematical treatment are referred to any of the standard works on applied mechanics.

APPLICATION TO SHELL RIVETS.

In the case of a ship, we have a beam very similar to the one used as an illustration in the beginning of this paper. We considered there a beam formed of several layers of wood, held together and made to work as one by spikes; and we discovered that there was a shearing stress set up in the

spikes. We may consider the ship as a beam wherein each strake of shell plating corresponds to one of the layers of wood in our illustration, and it will be seen at once that the rivets in the seams perform the same function as the spikes.

Now the shearing force as determined by equation (4) exists throughout the cross section of the ship, but is resisted by less material at the seams than elsewhere, and hence it is necessary to investigate only the conditions in the seam and the stress to which the seam rivets are subjected.

In equation (4), if all the values are so taken as to give the result in terms of shearing stress per square inch, we can replace the term b by the area of material per inch of length which is subjected to the shear. Thus if

a = area of one rivet,

and

p = pitch in inches,

the area per inch of length is $\frac{a}{p}$ for each row of rivets, and for a single riveted seam, observing that the resistance is divided between the two sides of the ship, we have

$$f = \frac{m S p}{2 I a} \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

for a double riveted seam,

$$f = \frac{m S p}{4 I a} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

for a treble riveted seam,

$$f = \frac{m S p}{6 I a} \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

and so on.

Two points are to be observed: First, these formulæ are based entirely on the rivets, assuming that they are the weakest part of the joint, and if the plate efficiency is less than the rivet efficiency (that is, if the area of the plate remaining between the holes for the rivets in the outer rows is less than the area of all the rivets, which is very likely to be the case for a treble riveted seam), the critical stress will be that in the plate, not that in the rivets, and its value can be obtained from the formula by increasing f in the ratio of the plate area to the rivet area, or by substituting

$$\left(t - \frac{d}{p}\right) \text{ for } \frac{a}{p}$$

in the formula (5), where t is the thickness of the plating in inches and d is the diameter of the rivet. Secondly, no account is taken of any longitudinal bulkheads which may bear a portion of the shear; their effect is better obtained by carefully studying the particular conditions, and modifying the calculations accordingly, than by any attempt to include such complications in the general formula, especially as the presence of such bulkheads generally removes all doubt as to the strength of the shell riveting, and renders any calculation unnecessary.

In applying the formula, it must be observed that the moment of inertia and the statical moment are not those for the midship section used in the ordinary strength calculation, but must be found for the section of the vessel which comes at the point of maximum shear. There are always two such points, and they come not far from the quarter length of the vessel from each end. A glance at Fig. 2 will show this. In this figure are shown the weight and buoyancy curves for a vessel, and the corresponding shearing force curve, the ordinates of which represent the difference between the weight and buoyancy from point to point. The maxima occur at S and S', and these are the values to be used in the equation, the moment of inertia and the statical moment of the vessel's cross section being calculated for these points.

As the amount of the vertical shearing forces have generally been computed incidentally to the standard strength calculation, the curve of shearing forces is available. The additional labor necessary to find the longitudinal shear consists, therefore, in making the inertia calculation for the section where the maximum shear occurs. In many cases, however, a fairly close approximation can be had without calculating this section, by taking advantage of the inertia calculation for the midship section, which has presumably been made for the standard strength calculation. Provided the general structure of the vessel does not change materially in type between the midship section and the point of maximum shear, the ratio of the statical moment to the moment of inertia will remain practically constant. The value of $\frac{m}{I}$ can therefore be obtained from the midship section, and used in equations (5), (6) and (7) for an approximate result. Of course, if this approximation shows the riveting to be in a critical condition the exact method should be resorted to.

As regards the statical moment to be used, it will be remembered that in making an inertia calculation it is necessary to calculate the statical moment of all the items above an assumed axis, and also the statical moment of all the items below that axis, and to find the location of the true axis by making these two moments equal, in order that the axis may pass through the center of gravity. It is one of these two equal moments which is to be used in our formula. If the seam we are investigating does not come right at the neutral axis, the moment of the intervening material is, of course, to be subtracted.

Some caution is necessary with reference to the units used. The equations have been based upon the assumption that all the measurements have been in inches. In the inertia calculation, however, it is customary to measure the areas in square inches and the lever arms in feet, so that the statical moment is obtained in terms of square inches times feet, and the moment of inertia in terms of square inches times feet squared. If this practice is followed the values of the statical moment and the moment of inertia must be multiplied by 12 and 144, respectively, in order that they may be in terms of inches only. The same thing may obviously be accomplished by dividing the value of $\frac{m}{I}$ by 12.

If the calculations indicate that the riveting is deficient, the question arises as to how the weakness may be remedied. The first thought is to add another row of rivets. It was mentioned above, however, that the plate may in some cases be weaker than the rivets, in which case merely to add to the number of rivets would not increase the strength of the joint. Or, the relative strength of the punched-out plate and the rivets may be such that, while the plate is slightly stronger than the rivets, the addition of another row of rivets would make the rivets much stronger than the plate, and the advantage of the added rivets would be largely, if not altogether, lost. In either case, it is necessary to find some other means of strengthening the joint.

Since watertight spacing of rivets must be maintained along the calking edge, there are two methods open to use. We can retain the lapped joint and fit a supplementary strap on the inside, or we can make the joint flush, with a regular seam strap on the inside. Either of these methods allows us to space the outer rivets in the strap widely, and so maintain the plate efficiency. If the flush seam is adopted, care must be taken to make the seam strap thick enough, since its strength will be that of the net area through the watertight

spaced rivets. Of course, neither expedient has to be employed throughout the entire length of the vessel, but only for such a distance as the magnitude of the shearing forces renders necessary.

THE VESTIBULE SOCIAL HALL ON THE CORCOVADO.

New German-Built Steamers for Brazil.

BY MAX A. R. BRUNNER.

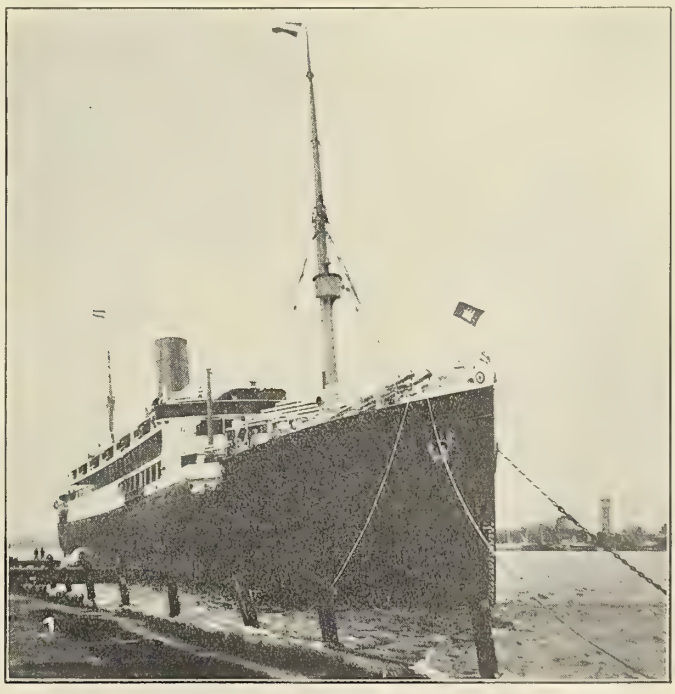
About a year ago the Hamburg-American Co. considerably improved its service to La Plata countries by installing the luxurious and fast boats *König Friedrich August* and *König Wilhelm II.* A further development is taking place this year, with two new ships, the *Corcovado* and *Ypiranga,* of which the former was ready for service in March, the other in July. These vessels surpass all others engaged in the Brazilian service, both as regards size and luxurious equipment of the passenger rooms. Each has a capacity of 7,800 gross register tons.

In both ships the experiences made on tropical cruises have been carefully utilized, so that they are specially adapted to the requirements in those zones. The cabins are very roomy and airy, are located in the high deck structures, well ventilated and furnished. All have their windows outside, there being no inner cabins on these boats. A novelty for the Brazil service is the large number of one-bed rooms. There are also cabins-de-luxe, consisting of parlor, sleeping room and bath. Each vessel has accommodations for about one hundred first class passengers. All staterooms are equipped with electric light, and many with telephones. White has been chosen as color for walls, ceilings, etc. This applies also to the general rooms.

The dining saloon is specially worth mentioning. It contains tables for five to eight persons, which shows again the tendency of German companies to abolish the table d'hôte system. Excellent taste is shown in the arranging of colors of the furniture, carpets, curtains, etc., matching the bright gold and white of the ceiling and walls.

Each vessel possesses a large hall, located on the promenade-deck, a room with eight corners and four niches. It is decorated in Roman style. Abundance of light is furnished by six broad windows and the skylight; the latter contains colored glass panes which represent four beautiful Brazilian landscapes. In the center are writing tables, a large piano, comfortable lounges and sofas, etc. One of the niches contains a fire-place; the others are furnished with book stands, club chairs, rockers, etc. By this arrangement the hall is neither exclusively a ladies' room nor only for gentlemen, but a social hall for both. It will be much occupied when rain or unusual heat makes the decks unpleasant.

A smoking saloon is also provided, furnished and decorated in Flemish style. It has dark oak squares in the walls and floor, and is decorated with Delft paintings representing various sports. Antique leather chairs and sofas constitute its furniture. Each boat contains a children's room, equipped with pretty oak furniture; this serves as dining and playroom for the little ones.



THE CORCOVADO APPROACHING COMPLETION.

Stairways, corridors, decks, etc., are very roomy and airy to render the stay on board pleasant, even in those very hot zones. The *Corcovado* has in the stairway vestibule a large painting representing the high mountain Corcovado, near Rio de Janeiro.

Each vessel has accommodation for about 1,200 steerage passengers.

THE FIRST CABIN SMOKING ROOM ON THE BRAZILIAN STEAMSHIP CORCOVADO.

NOTES ON NAVAL SCIENCE TOPICS.

BY ARTHUR R. LIDDELL.

STABILITY.

Much has been written within the last thirty years about stability, and many methods have been put forward for determining the righting moments of vessels inclined to different angles under given conditions of wind, weather, and loading. There is a general tendency, however, to avoid the subject, first, because the methods for dealing with it entail time and labor, and, secondly, because clear ideas as to the degree of righting tendency necessary for a vessel under the various conditions of her service have yet to be formed and embodied in rules for general guidance. As a rapid method of approximately determining the stability of vessels of full form, the following may be proposed:

The conditions in such a vessel are approximately the same as those of an equivalent rectangular block, such as that shown in section in Fig. 1. The curve of levers of the latter may be quickly obtained as follows:

Assume a loadline *AB* (Fig 2) and incline the rectangle until the surface of the water successively assumes the positions *CD, EF, GH,* and *IK* with reference to it. With the exceptions of those at *EF,* the points in which these position lines cut the outlines of the figure may at once be determined. In regard to *EF,* if the rectangle *MB* be *a* times *MN,* and

$$\frac{ME}{MO} = \frac{ME}{MD}, \text{ then } \frac{ME}{MG} = \sqrt{2a}$$

and the centers of the figures *MEF* and *EGNDF* lie on the diagonal *MN.* Their distances from the middle of the rectangle, being in inverse proportion to the areas of the part in question, can also be readily determined.

Referring again to Fig. 2, the centers of the immersed parts of the rectangle at the different inclinations may now be found by simple methods which need not be here elaborated. The line XX_1X_4 in Fig. 3 is the locus of these.

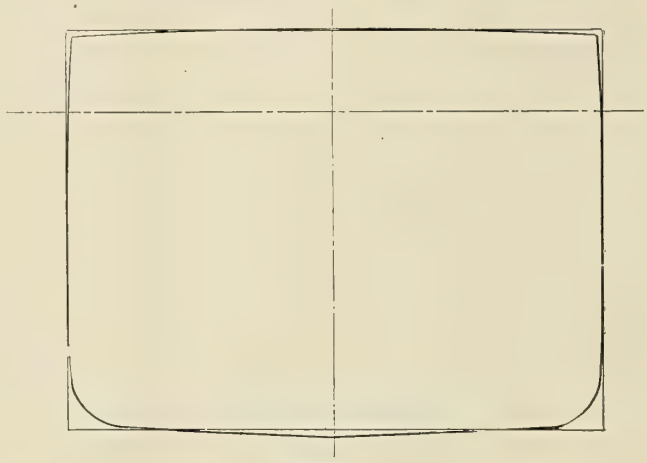


FIG. 1.

We may now assume the side *ND* to be produced upwards, so that the line *GH* would, at the same inclination and for the same immersed area, take up the position G_2H_2. For this condition of things the new center is at Y_3, and a further center Y_4 may be obtained, say for a triangle *PQN* of the same area, but of twice the height of H_2G_2N. A new locus, diverging from the old one, may now be drawn through the points Y, Y_3, Y_4, which would satisfy the condition of the vessel's having erections of unbounded height extending all fore and aft. As a matter of fact the erections will be of a certain limited height, and will extend over a portion only of the length.

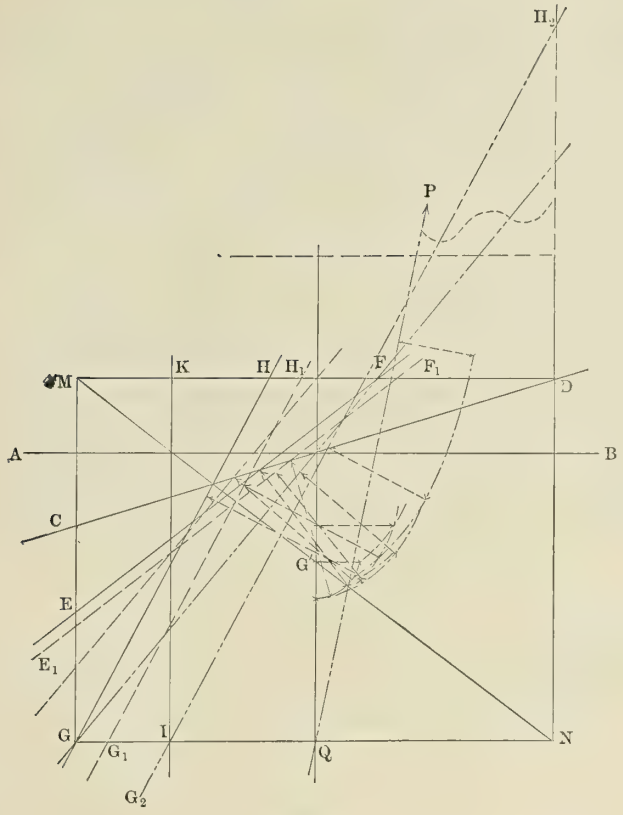


FIG. 2.

A locus XZZ_4, in a position intermediate between those of XX_1X_4 and XYY_4, may now be sketched in, as shown by the dotted line in the figure, and, if necessary, checked and corrected in accordance with the real heights and longitudinal distribution of the erections, whereby the latter may be taken as having the breadth of the rectangular section and their own actual longitudinal positions. This correction, though not a very difficult one, forms the most troublesome part of the rectangular method.

Fig. 4 shows a curve of levers measured from Fig. 3 on the assumption that the center of gravity of the vessel coincides with her center of buoyancy when she is upright.

The probable position of the actual center of gravity may, in the meantime, have been assumed or approximately determined, say by the method sketched in the August (1907) number of this journal. Assume that it takes the position G, in Fig. 3. Since the heights of the centers of gravity above, or their depths below, the metacenter become direct leverages at an inclination of 90 degrees, the height CG may be set down from the 90 degrees spot of the lever curve in Fig. 5.

Now the curve of levers (or alteration of levers), due to an alteration in the height of the center of gravity of a vessel, is simply a curve of sines. The lever for an inclination of 30 degrees is, for instance, equal to CG sin 30 degrees, or $CG \times \frac{1}{2}$, and a distance of $\frac{1}{2}$ CG may then at once be set down from the original lever curve at the station for 30 degrees. A curve of sines may, in fact, be quickly set off from the original curve, and the new line will be the curve of levers corresponding with the G position of the center of gravity. In similar manner, additional lever curves may readily be drawn corresponding with other positions of the center of gravity, and the designer will be in a position to judge fairly enough as to the stability characteristics pertaining to his proposed model.

For sketching out curves for different positions of the center of gravity, it is convenient to remember the following approximate figures:

Sin 7½° = 0.13	Sin 45° = 0.71
" 11½° = 0.2	" 64¼° = 0.9
" 30° = 0.5	" 90° = 1.0

The characteristics of the lever curves of vessels of finer form will be less like those of their equivalent rectangles, but a little experience with the method, on the part of the designer, will enable him to sketch in the necessary corrections. An example of this, for a fine-lined sailing vessel, is given in Fig. 5, which shows lever curves for GM heights of 0 and 3 feet, respectively. The displacement of the vessel was 3,100

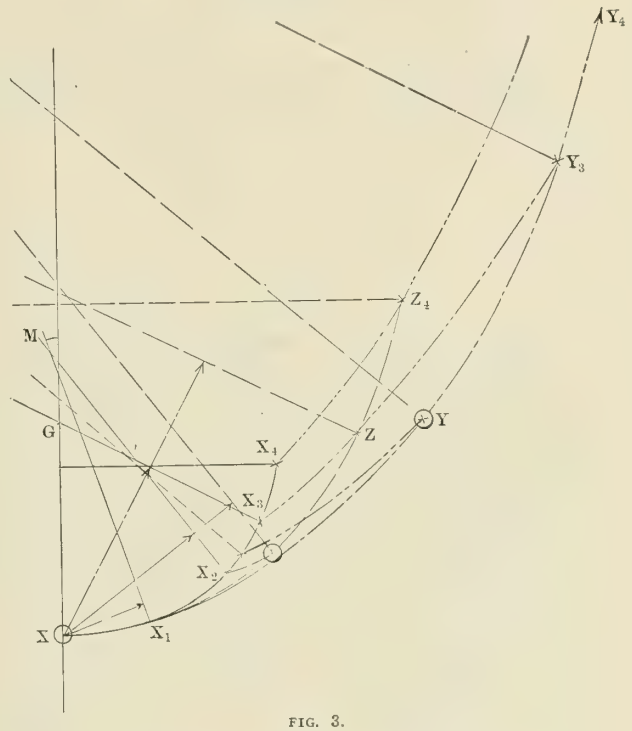


FIG. 3.

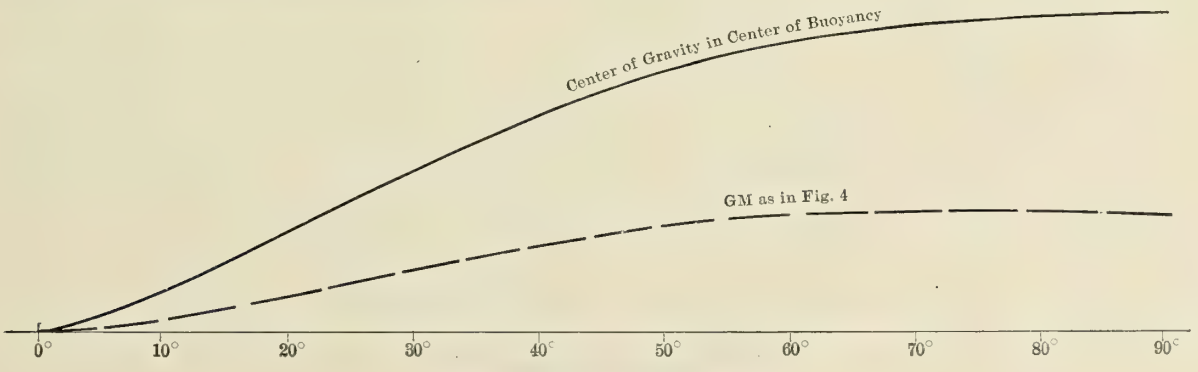


FIG. 4.—CURVES OF LEVERS FROM FIG. 3.

tons, and her block coefficient 0.66. The full lines are obtained by the ordinary methods, and the dotted ones by the rectangular one above described. The principal differences are that due to the tumble home of the vessel's side, which acts in lowering the curve between the inclinations of 20 degrees and 65 degrees, and that caused by the distribution towards the deck of the displacement of the fine ends.

For many purposes the approximate curves would suffice without correction, even for this rather extreme case, while for vessels of box-like form, with nearly vertical sides, the differences are very much smaller. If, for instance, the middle body of the above vessel were increased in length until her block coefficient became about 0.8, the differences between the lever curves as given by the two methods might be one-third of those shown in the figure.

The question as to how much stability a vessel ought to have must be answered according to her class and service. As mentioned in the August number, the *GM* height required for a sailing vessel has been put at from 3 feet to 3 feet 6 inches for all classes of sailing vessels. Provided the proportion of breadth to draft, or of area of load waterline to displacement, lies within ordinary limits, this height is approximately correct. In the case of vessels that are broad in relation to their depth, it is necessary to increase the *GM* height, because the righting levers, though relatively large at the smaller angles of heel, are relatively small at the larger ones. This state of things may become dangerous in two ways: A lull in the wind, followed by a squall, while a vessel is sailing, may cause her to capsize; for while she is apparently standing up well to the steady inclining force applied to her sails, she may have a comparatively small reserve of righting moment. Again, the waves have much more power over a vessel when her proportion of depth to breadth is small than when it is large; and, should she meet with waves of a period equal to her own, as is always possible, she may roll up to dangerous angles in a very short time. That this must be so will be clear when it is remembered that the influence of the waves increases with the square of the breadth, while, within ordinary limits, the inertia of the vessel varies approximately with the product of her breadth by her depth.

A vessel of this kind should have a curve of levers which cuts the base-line at not less than about 60 degrees. She will, under these conditions, be uncomfortable, but comfort must within reasonable limits be sacrificed to safety. The height of the center of gravity *G*, which must be aimed at in the distribution of the weights of the vessel, may be found as follows:

Assuming the curve of levers for zero *GM* height to have been obtained, say by the approximate method described, the distance below the base-line of a point in the curve at 60 degrees, divided by the sine of this angle, will give the *GM* height required, and this, multiplied by the sine of any angle between 0 degrees and 90 degrees, will give an offset for the new curve. The latter curve must be set off from the old one, and, of course, cuts the base-line at 60 degrees, as required.

For a cargo steamer the problem is a different one. Approximately constant inclining forces, such as that of the wind on the sails, are here of small account. The all-important conditions are that the vessel should stand upright in smooth water under all conditions of loading, and that she be safe against being rolled over in a seaway.

As regards the first of these, the vessel should, under ordinary conditions, have positive levers up to inclinations of about 60 degrees. In the most unfavorable state of her bunkers, and with the largest amount of free water in bilges and double bottom that is likely to get there on service, she must have a *GM* height of at least 6 inches. In vessels in which breadth is large in proportion to depth, a range of positive stability up to 60 degrees may not be attainable without the accompaniment of excessive stability at the small angles. In such cases the range may have to be reduced, but the special circumstances should then be duly weighed.

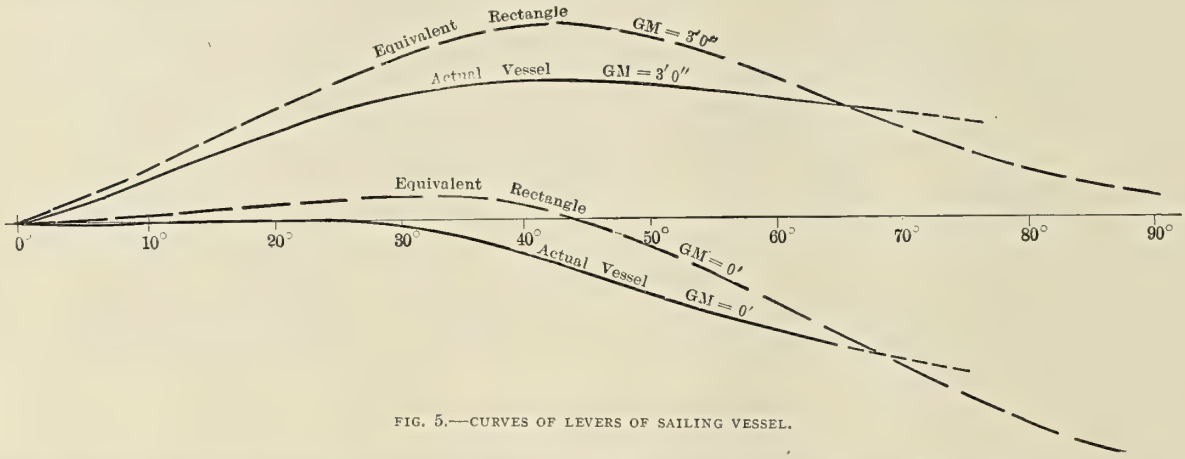


FIG. 5.—CURVES OF LEVERS OF SAILING VESSEL.

In small vessels it would be well to provide for a *GM* of about 12 to 15 inches, since they are at any time liable to meet with waves of the same period as their own, which may set up excessive rolling. Very large vessels with small *GM* height will probably have longer periods of rolling than the periods of any waves they are likely to encounter.

Low *GM* heights, such as those mentioned, make for steadiness and comfort at sea, but the vessels in which they occur must neither get into collision nor become leaky, and their captains must not attempt to run up broad water ballast tanks. Larger *GM* heights insure greater safety; but, after all, such safety is only a matter of degree, and the seagoing condition of a vessel is simply a balance of dangers and advantages. Neither absolute safety nor absolute comfort is as yet attainable.

Increased attention is beginning to be given to methods for the prevention of rolling, and it seems not unlikely that one or other of these may soon bring success in a commercial as well as in an engineering sense. Meanwhile the excessive rolling that results when the period of the waves synchronizes with that of the vessel may be mitigated, either by a change of the latter's course or by the expedient of partly filling or partly emptying a broad water ballast tank. Maneuvering with the water ballast tanks at sea in this way may, however, prove a dangerous experiment, and it is one that should be resorted to only in cases of emergency, and by men who are alive to its possible effects.

A small quantity of water introduced into a tank or bilge, or a small quantity pumped out of a full tank, may suffice to produce a fluid surface that does not regularly accompany the

motions of the vessel. Her period at once becomes altered, and, synchronism being thus destroyed, her rolling inclinations will be moderated. But it is a well-known rule that water ballast tanks should neither be filled nor emptied at sea, and to do so in the case of a tender vessel is, indeed, likely to prove disastrous. In a vessel of large stability, however, commanded by a man who knows what he is about, the operation above described might sometimes be resorted to with advantage.

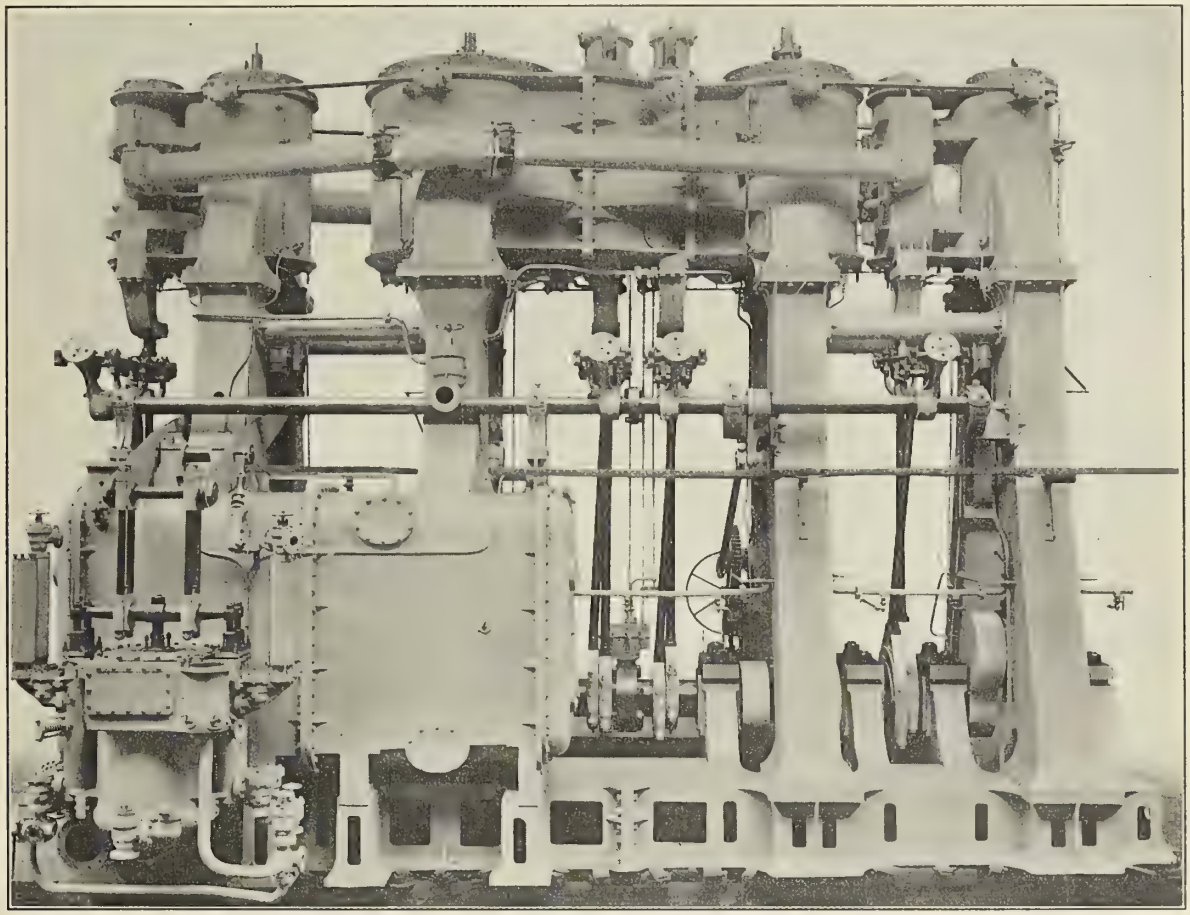
Assuming perfect synchronism of periods, a vessel will heel further over with each successive wave, until either she capsizes or some new influence intervenes to alter her period. An influence of this kind comes into play when her gunwale becomes submerged and water gets on deck. The rolling motion is thus damped for awhile, and then gradually increases until the next dip takes place. But such a shipping of water may in itself be highly dangerous, and is in all cases objectionable.

A danger that is not always recognized lies in the wells of quarterdeck steamers, and in similar catch-water spaces on vessels of other types. Assume that a wave has filled the well to such a height that the water is free to preserve an approximately level surface while the vessel rolls. The effect of this is more than equivalent to that of the loss of a piece of the plane of flotation of like shape and size, since the center of gravity is at the same time raised. The height of the metacenter above the center of buoyancy being equal to

$$\frac{\text{Moment of inertia of waterplane}}{\text{Displacement of vessel}},$$

it will be seen that the loss may be a heavy one. It will perhaps be urged that the water runs off through the freeing ports; but, when these have the largest area possible, the time required to drain the well is very considerable, even if a second wave does not replenish the supply.

The perfection, which may now be hoped for, of some device for preventing or damping rolling, will shelve some of the most awkward problems of stability. Apart from the question of passengers' comfort, vessels provided with such devices will be less strained in a seaway, and less dependent upon wind and weather, and the safety of their crews will be promoted in a high degree. Whatever the device adopted, it will cost a certain amount of money and take up some space and weight, but it is highly probable that these disadvantages will be more than counterbalanced by reductions in coal bills, increase of speed in rough weather, and comparative absence of wear and tear in the structure.



BACK VIEW OF THE MAIN ENGINE OF THE NORSE PRINCE, SHOWING CONDENSER AND AIR AND CIRCULATING PUMPS.

The Steamship Norse Prince.

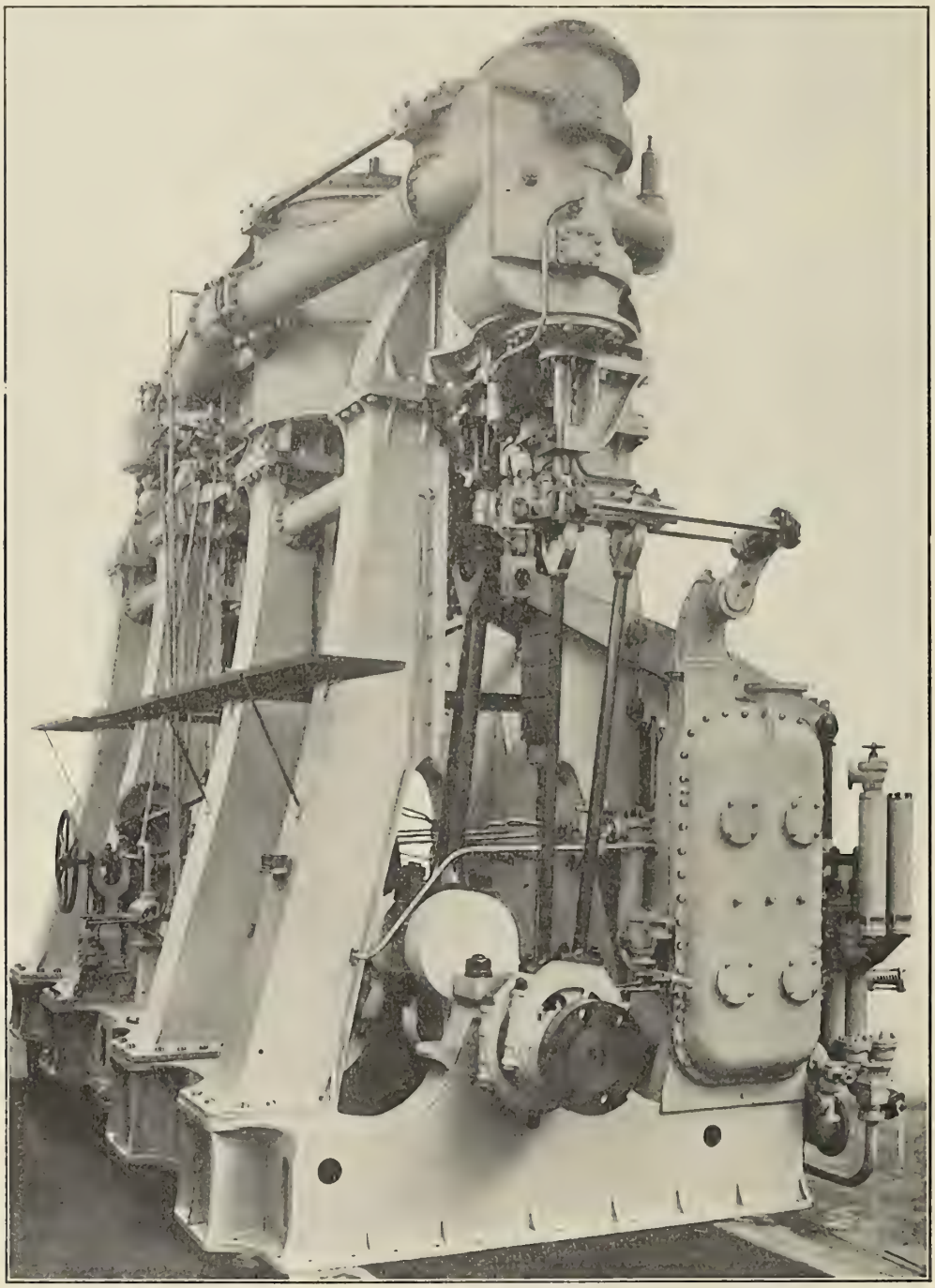
There was launched on Sept. 10, 1907, from the yard of Palmer's Shipbuilding & Iron Company, Ltd., Jarrow, a handsomely modeled steel screw steamer of the following dimensions:

Length between perpendiculars	420 feet
Breadth molded	54 feet
Depth molded	31 feet 1 inch

The vessel has been built of steel, with the usual parts of iron, to the highest class in Lloyd's Register; she is of the three-deck type, having a continuous shelter deck all fore and aft. The main, upper and shelter decks are of steel all fore and aft, the latter being wood sheathed. Accommodation is fitted in steel deckhouses on the shelter deck amidships for captain, officers and twenty-four first class passengers; the crew being located under the shelter deck forward. The shelter deck and 'tween decks are fitted with side lights and

ventilation necessary to enable the vessel to carry cattle or emigrants.

The vessel is divided by eight watertight steel bulkheads. She is fitted with cellular double bottom for water ballast, extending all fore and aft except under the boilers; water ballast being also contained in a deep tank abaft the engine room, vided with steam by three boilers, 14 feet 9 inches in diameter by 11 feet 9 inches long, at 220 pounds per square inch working pressure, under Howden's forced draft. Generally, the vessel, which will load over 9,000 tons deadweight, is equipped with all modern conveniences to fit her as a first class cargo and passenger steamer.



THE MAIN ENGINE OF THE STEAMSHIP NORSE PRINCE, SHOWING HANDLING GEAR.

extending to the main deck and in fore and after peaks. The ship is rigged as a two-masted fore-and-aft schooner, and a very complete system of winches and derricks is provided for expeditiously loading and discharging. Electric light is fitted throughout, and a direct steam windlass is placed forward for working the anchors; the steam steering gear being fitted in wheelhouse aft, and geared direct onto the tiller.

There is one quadruple expansion engine, having cylinders 24½, 35, 51 and 74 inches in diameter by 51-inch stroke, pro-

The North Dakota, the third of the American all-big-gun battleships, is to be launched by the Fore River Shipbuilding Company on Nov. 10, being about 60 percent completed. The normal displacement is 20,000 tons, on a draft of 26 feet 11 inches, the length on waterline being 510 feet, and beam 85 feet molded. She carries ten 12-inch guns in pairs in five turrets on the center line, all bearing on one broadside. Propulsion is by twin screws, operated by Curtis turbines of 25,000 horsepower.

THE HEATING AND VENTILATING OF SHIPS.

BY SYDNEY F. WALKER, M. I. E. E.

THE HEATING BY ELECTRICITY OF A LARGE PASSENGER STEAMER.

The writer has outlined a scheme for heating a large passenger steamer throughout by electrical apparatus, and he has selected for the purpose the North German Lloyd steamer *Kaiser Wilhelm der Grosse,* which has a displacement of 20,000 tons, carries a crew of 450, and has accommodation for 1,500 passengers—first, second and third class. Her dimensions are: Length, 625 feet; breadth, 66 feet; depth, 43 feet. It will be understood that a scheme of the kind can be only approximate. As engineers know well, the figures for each case, in engineering practice, have to be worked out by themselves, and the present writer has not the whole of the necessary figures before him to enable him to make an exact estimate. The figures will, however, he believes, be sufficiently accurate to show what can be done by electrical heating apparatus, and what it is likely to cost. The writer has also taken into account in the heating appliances only the passenger accommodation and its accessories.

The *Kaiser Wilhelm der Grosse* has:

A first class dining saloon, measuring 110 by 65 feet, placed amidships on the main deck; a second class dining saloon, measuring 50 by 55 feet, aft on the main deck; a children's saloon, measuring 44 by 26 feet, aft on the main deck; a children's dining saloon, measuring 45 by 30 feet, forward on the main deck; an auxiliary second class dining saloon, 50 by 15 feet, aft on the upper deck; a first class drawing room, 32 by 30 feet, amidships on the promenade deck; a second class drawing room, 25 by 20 feet, aft on the upper deck; a first class smoke room, 33 by 40 feet, forward on the promenade deck; a second class smoke room, 38 by 26 feet, aft on the promenade deck; a reading room, 30 by 20 feet, forward on the promenade deck.

It has also accommodation for third class passengers on the lower deck totaling 200 by 34 feet; also for stewards for third class passengers on the lower deck totaling 120 by 20 feet; accommodation for third class passengers on the main deck, 42 by 40 feet; accommodation for attendants on the main deck, 40 by 20 feet.

In addition there are some 260 spaces to be heated, comprising staterooms, hospitals, kitchens, pantries, lavatories, etc., and there are the usual vestibules to the saloon, in which the stairs leading from one deck to the other in the passenger departments are fixed.

The height between decks for the promenade, upper and main decks, is 8 feet, and that of the lower deck is 7 feet. There is the usual orlop deck, but it is occupied mainly by boilers, coal bunkers, cargo, baggage, chain lockers, etc.

The writer has divided the staterooms and similar spaces into two sizes to simplify the calculations. One lot, of which he makes out that there are approximately 100, measure 10 by 10 feet by 8 feet high. The smaller ones, of which there are about 160, measure 8 by 6 feet by 8 feet high.

In calculating the heating apparatus required and the quantity of current to furnish the necessary heat, the writer has worked to the same figures as were used in explaining how to calculate the quantity of heat that must be provided by any heating apparatus to make up for the heat lost by passing out through the sides of the ship, the bulkheads of cabins, saloons, etc., and the decks above and below. The problem, it will be seen, is similar to that which the refrigeration engineer has to deal with. In the case of cold storage the problem is to carry off the heat that passes in through the walls, decks, etc., of the cold chamber. In the present case the problem is to deliver heat to the rooms to be warmed, to make up for that which has passed *out* through the walls, decks, etc.

To estimate the quantity of heat that must be delivered by any heating appliance, the quantity of heat passing out of the room to be warmed must be estimated by taking the surfaces through which heat can escape, the quantity of heat escaping per square foot, and the difference of temperature between the inside and the outside. The rate of passage outwards of heat the writer has taken at the figure given in a previous article, 0.5 British thermal unit per hour per square foot per degree F. difference of temperature between the inside and the outside; with one exception, that of the skylights of the first class dining saloon and the first class drawing room. The rate at which heat passes through glass is very much higher than that at which it passes through wood, and it has been assumed, in all the calculations, that wood is the substance through which the heat from the saloons, state rooms, etc., has to escape, except in the case of the skylights mentioned.

The writer has also taken the figures mentioned in a previous section, viz.: a minimum temperature of 30 degrees F. outside the ship and a temperature to be maintained in the rooms to be warmed of 70 degrees F. In a great deal of the passenger service the minimum temperature mentioned, 30 degrees F., will not be reached. Probably a temperature of 40 degrees F. would be more like the average, but even crossing the Atlantic in winter very much lower temperatures are met with. The calculation is intended as a guide only, and must be altered to suit the temperatures, and it is a very simple matter to do so. Thus, the extreme range of temperature taken in the writer's calculations is a difference of 40 degrees F. For ships which meet temperatures of 20 degrees F. an addition of 25 percent to the heat required to be delivered will be necessary, and to the heat-furnishing appliance. To ships meeting temperatures of 10 degrees F. an addition of 50 percent to the writer's figures will be required. For the average minimum of 40 degrees F., which will probably meet the case of a large number of ships, the writer's figures may be reduced by 25 percent.

It should also be noted that the writer has taken 70 degrees F. as his standard of temperature within the rooms to be warmed, this being the temperature to which Americans are accustomed. Europeans are accustomed to take standard temperatures of 60 degrees, or at most 65 degrees F. For those ships where a standard of 60 degrees F. would be sufficient for living places the writer's figures may again be reduced by 25 percent.

In drawing his estimate, also, the writer has assumed that the heating appliances would be placed in the best position for distributing the heat to the best advantage. On board ship there is not the same trouble as on shore with chimneys, except in those saloons that are furnished with grates, fires and so on, and therefore there is not the danger of the heat liberated by the heating appliance being carried off up the chimney. Ventilation, of course, must be provided, and, as already indicated, the best method is to place the heating appliance in the path of the ventilating air current, where it enters the room to be warmed. Where there is no special ventilating arrangement the heating appliances should be placed as far as possible in the line of the natural ventilating air current, the air which comes in under doors and by other openings as it enters the room.

There are two or three important points that should be noted in connection with the calculations for the size of heating appliance required and the quantity of current. Thus, of the two sizes of staterooms, the larger measuring 10 by 10 feet by 8 feet, or 800 cubic feet, and the smaller 8 by 6 feet by 8 feet, or 384 cubic feet, say 400 cubic feet in round figures; owing to the much larger surface in the 10-foot rooms above that exposed in the 8-foot rooms, the amount of heat passing out through the walls, etc., and the amount of heat therefore required to be delivered to the air of the rooms to make it up, is enormously larger for the large rooms than for the smaller, as will be explained.

In the present case, also, the first class dining room and drawing room are exceptionally well placed, as far as the requirement of heating appliances is concerned, because they lie between two funnels. The funnels, of course, are protected on the outside, so that the passage of heat outwards is a minimum, but the writer has assumed that a certain quantity of heat does pass from the funnel casing into the dining and drawing rooms. It is a case of heat passing into these rooms in place of passing out of them, and the quantity of heat that has to be delivered by any heating apparatus to these rooms is lessened by the heat delivered from the funnels. In estimating the heat required for the dining saloon the writer has taken the heat passing out through the ship's side, the decks above and below and the skylight, and has subtracted from it the heat he estimates will pass in from the funnel casing. In the case also of the alleyways, as the engine-room funnel and stoke-hold casings will line the alleyways for a large portion of the length of the ship, and as a considerable amount of heat must pass from them into the alleyways, the writer has assumed that the temperature of the air in the alleyways will be raised 10 degrees above that of the air outside during cold weather.

APPARATUS ESTIMATED TO BE REQUIRED FOR HEATING THE DIFFERENT SALOONS, STATE CABINS, ETC.

The electrical heating appliances, it was explained, are made in various sizes to absorb from 200 watts up to 6,000 watts. These heating appliances, it will be remembered, are divided into two distinct varieties, those in which the long incandescent electric lamps are employed and those in which non-luminous resistances are employed. The lamps are usually arranged to absorb 250 watts each, though the Prometheus Company, of Great Britain, has recently introduced lamps taking 350 watts, and burning at a red heat in place of a yellow heat. The 250-watt lamp, however, forms a very convenient standard, particularly as it is made up into appliances carrying 2, 3 and 4 lamps, and in the following estimate the writer gives the figures in watts required and in lamps of 250 watts each:

The first class dining saloon requires the expenditure of 4,500 watts, or 18 lamps. The first class drawing room requires the expenditure of 1,400 watts, or 6 lamps. The second class dining room requires 3,500 watts, or 14 lamps. The children's dining room requires 3,000 watts, or 12 lamps. The children's saloon requires 2,500 watts, or 10 lamps. The auxiliary second class saloon requires 1,250 watts, or 5 lamps. The first class smoke room requires 2,000 watts, or 8 lamps. The second class smoke room requires 1,500 watts, or 6 lamps. The reading room requires the expenditure of 1,000 watts, or 4 lamps. The third class passengers' quarters, stewards, attendants, etc., require 6,000 watts, or 24 lamps.

The large staterooms, the writer makes out, require 2,500 watts, or 10 lamps. and the smaller staterooms 250 watts, or even less, or 1 lamp. Taking the number of the larger staterooms at 100, this means 1,000 lamps in addition, and the number of the smaller rooms at 160 means 160 more lamps. In addition to the above there are the vestibules mentioned.

The total number of 250-watt lamps given in the above list is 1,267, and it will therefore be wise to allow for a total, to cover all contingencies, of 1,500 lamps of 250 watts, or for a current of 375 kilowatts.

The above figures are for the minimum outside temperature mentioned, 30 degrees F. If a temperature of 20 degrees F. has to be provided for 1,875 lamps, or say a current of 470 kilowatts, would be required. With a temperature of 10 degrees F., 2,250 lamps and a current of 565 kilowatts would be required. With a temperature of 30 degrees below zero F., or 100 degrees F. difference between the outside and the rooms to be warmed, 3,750 lamps would be required, and a plant capable of furnishing 940 kilowatts, or about 1,250 horsepower.

THE COST OF FURNISHING THE HEAT REQUIRED.

So far as the writer has been able to ascertain, no figures have yet been accurately taken out giving the cost of generating current on board ship. With electric lighting as an auxiliary the steam required has gone in with other auxiliaries. On the other hand, it is often not the rule to condense the steam used by auxiliaries, and the consumption of electric light engines is taken at about 40 pounds of steam per kilowatt-hour. But it must be remembered that the steam is being generated under the most economical conditions. The steamer crossing the Atlantic, or at sea for any number of days, under any possible conditions, providing she is continuously steaming, is generating steam under the most favorable conditions known to the engineer. There are practically no stand-by losses, such as send up the fuel cost so much on shore.

On shore, it will be remembered, lighting is required only for a certain number of hours during the day, and power even is required only for a certain number of hours, and there are portions of the day, both with lighting and with such power services as that for tramways and suburban railways, when the quantity of current is very high indeed for a short time, dropping to something very small between those times. This means that either boiler furnaces have to be banked or they have to be let out and relighted. In either case it means the consumption of a considerable quantity of fuel not required in steamship work. Further, the oil and petty stores and other things required for electrical generating plant on board ship, are part only of a large whole, and therefore should be obtained more economically, providing proper care is used in issuing stores, than where the plant, often a comparatively small one, has to buy everything specially for its own use.

The attendance, also, in the case of a shipboard electricity generating plant, should be smaller than for a similar plant on shore. Boiler attendance goes in with that of the main engines. Even a very large electricity generating plant will not make much appreciable difference to the stoke-hold labor of a ship of 20 000 tons and of 27,000 horsepower, such as the *Kaiser Wilhelm der Grosse*. Attendance, in fact, is resolved into that of the electrician on watch, with possibly an assistant to oil, and the electrician to look after the apparatus in use, to make good little breakages, keep switches right, etc.

The writer thinks that he will not err on the side of optimism if he takes the cost of electricity at 0.5d. (one cent) per kilowatt-hour (Board of Trade unit, as it is called on shore in the United Kingdom). In America there are many electrical generating plants, generating current for railways and other purposes, in which the cost is very much less than the figure taken above, and there are cases even in the United Kingdom of electrical generating plants at collieries and in other works where the cost of generation is much less than 0.5d. If the above figure is taken, therefore, any estimates founded upon it should be fairly safe.

The heating appliances detailed for the different saloons, etc., total up to 107 lamps of 250 watts each. These heating appliances will probably be required from 8 A. M. to midnight, or say 16 hours, and in addition a certain number of the smaller staterooms, and possibly a few of the larger ones, will probably be required during the same hours, and therefore it will probably be approximately correct to assume 240 lamps of 250 watts each as being employed for 16 hours per day. This means 60 kilowatts for 16 hours, or 960 kilowatt-hours per day. The writer estimates that probably 60 lamps of 250 watts will be required for the full 24 hours; this equals 15 kilowatts for 24 hours, or 360 kilowatt-hours. Of the remainder, the staterooms and other places will be required for probably 4 hours during the day. In addition there will be some 1,200 lamps of 250 kilowatts, or the equivalent, which will probably be required for 4 hours out of the 24.

This means 300 kilowatts for 4 hours, and equals 1,200 kilowatt-hours. Adding these together, the total is 2,520 kilowatt-hours per day.

This would cost, at 0.5d. per kilowatt, £5.5.0 ($25.55) per day, or say £31.10.0 ($153.30) for a six-day passage across the Atlantic. If 20 degrees F. is taken as the minimum temperature the total cost will be £39.7.6 ($191.62) for the trip; if 10 degrees F. is the minimum, £47.5.0 ($229.95), and for lower figures in proportion. For ships trading to very cold climates, and where economy is essential, where also the heating would be required for months together, the cost would probably be prohibitive in the great majority of ships. With a temperature of — 30 degrees F., or 100 degrees F. difference, the cost of heating would be £13.2.6. ($63.88) per day for a ship of the size of the *Kaiser Wilhelm der Grosse,* and with its crew and passengers. But as ships which go on whaling cruises do not carry passengers nor large crews, electric heating even then might be found economical, on account of its convenience, in some cases.

The heating can be carried out at less cost by the steam or hot water appliances that have been named, but as in so many other things the great convenience of the electrical method of distribution and of control more than counterbalances the increased cost, which, as will be seen, is but a very trifling sum, as against the whole cost of running a steamship of the size of the *Kaiser Wilhelm der Grosse* across the Atlantic. There are, of course, other points to be considered. Additional plant will be required to furnish the current, and it would probably not be safe to have less than 500 kilowatts for the special heating appliances in the case of the ship considered, and under the conditions named, and larger plant in proportion for the lower temperatures.

There is the question of finding room for a 500-kilowatt plant and the larger plant where required, though with turbo-generators this question is reduced to a minimum. There is also the question of the additional cables. Already the cable problem is a somewhat serious one in connection with lighting, and the addition to it of the requirements for heating will increase the trouble. There is no reason, however, that the heating current should not be taken off the lighting service, and, providing the conductors for the lighting service are properly divided up, as they always are in modern steamships, so that it is hardly possible for all the lights to be out at once, the trouble of the increased size of the conductor will not be so great. When a cable reaches a certain size a comparatively small addition in diameter gives it a considerably increased conducting power.

For the lower temperatures the cable question would be more serious. Everything, first cost and running cost, increases as lower and lower temperatures have to be provided for; but, again, in the case of the whaler, as the whole plant would be small, the matter need not be very serious.

Steam Turbines in the German Navy.

In no country has the marine turbine met with more bitter criticism or such a thoroughly hostile reception as in Germany. Unlike marine engineers in nearly all other countries, who tactfully reserved their judgments, the German naval authorities and the responsible officials of the large steamship lines condemned the entire system at the beginning, in no measured terms and on utterly insufficient grounds. It is a matter of some speculation, therefore, as to how some of these gentlemen can reconcile their consciences with the recent orders for turbine warships, or maintain their attitude in spite of the results obtained from certain naval vessels.

Almost simultaneously with the British Admiralty orders for the cruiser *Amethyst* and destroyer *Eden,* the German Government ordered a similar type of cruiser—the *Lübeck*—and a somewhat similar destroyer—*S125*—from the Vulkan and Schichau yards, respectively, in order to gain experience of the Parsons system. About the same time, the Hamburg-American Line had constructed at the Vulkan works the steamship *Kaiser** for their Hamburg to Heligoland service, which was fitted with Curtis turbines. Both the *Lübeck* and *S125* were subjected to comparative trials with sister ships, and each vessel was tried with several different sets of propellers. Some results from the *Lübeck* are given in Tables I. and II., while the figures for *S125,* and also for the larger destroyer by which she was succeeded—*G137*—are given in Table III. The *Lübeck* has four shafts.

Early in 1906, in spite of considerable criticism, a similar ship to the *Lübeck*—the *Stettin*—of slightly greater dimensions, was also ordered to be fitted with turbines, and yet a third of the series—the *Ersatz Comet*—was commenced early in 1907, at the works of Messrs. Blohm & Voss, at Hamburg. The small cruisers, however, ordered late in 1907, will be of a more experimental nature. The *Ersatz Greif,* building at Schichau's works at Dantzig, is being fitted with turbines of the Melms and Pfenniger type—a modification of the Parsons system whereby a shorter turbine is obtained. The two sister ships building at the Vulkan and the Germania Works will be fitted with Curtis and Zoelly turbines, respectively. It will be a matter of considerable interest to note how these different systems compare with one another.

The largest order placed, however, for turbine machinery is that for cruiser *F,* the German reply to the British *Inflexible* class. Details concerning this vessel are extremely meager, but her displacement of 19,500 tons is significant. There is little doubt that she will be constructed with all the secrecy and dispatch possible. She is to be ready before the end of 1909. No battleships have, as yet, been ordered with turbines, the four ships laid down early in 1907—the *Ersatz Sachsen* class—having the usual arrangement of triple screws and reciprocating engines.

In Table IV. are given the leading particulars of the vessels in the German navy fitted with turbines. No merchant steamers have as yet been built with Parsons turbines; of the two that have been constructed with turbine machinery, one has had Curtis and the other Zoelly turbines.

Table I.—Propellers.

Set	Propellers	Diameter	Pitch	Pitch Ratio	Projected Area	Ratio Projected Area / Disk
1	8 small	54.0″	54.0″	1.0	6.37 sq. ft.	0.4
2	4 large	66.9	58.8	0.88	9.5	0.39
3	4 mixed	(*a*) 62.9	56.4	0.896	12.9	0.6
		(*b*) 68.8	61.9	0.898	15.5	0.6

(*a*) One on each outer shaft. (*b*) One on each inner shaft.

Table II.—Speed and Power Results.

Propellers	Speed	Horsepower	Slip, percent	R.P.M.	Remarks
Set 1.....	22.37	13,705	25.11	672	
Set 2.....	22.39	13,029	25.97	623	Shallow water.
Set 2.....	23.16	14,158	26.5	...	Deep water.
Set 2a.....	22.56	13,573	15.51 for'd 22.67 aft.	601	Four screws of Set 1 forward, with four of Set 2 aft.
Set 3.....	22.55	13,879	25.79	625	

The displacements on these trials were between 3,200 and 3,300 tons. The horsepowers are *torsional* horsepowers, and were taken by the Föttinger torsion meter. The revolutions are the mean of the four shafts. Assuming the equivalent "indicated" horsepower to be 10 percent greater than the torsional horsepower, and the mean displacement to be 3,250

* See page 473, December, 1906.

tons, the Admiralty coefficients for these five sets of trials become 162.4, 171.2, 174.5, 168.1 and 164.2, respectively.

TABLE III.—TURBINE AND RECIPROCATING ENGINED DESTROYERS.

Vessel	Speed	Displacement	Horsepower	R.P.M.
(*a*) *S*120–124.....	27.5	390	6,850	390
*S*125.....	26.6	440	7,000†	875
*S*131.....	29.6	525	10,250	395
*G*137.....	33.09	570	13,000†	900

The *S* 120-125 class were completed in 1905, and the 131-137 vessels in 1907. (*a*) Sister ships to the turbine vessels, but fitted with reciprocating engines. †Estimated.

TABLE IV.—TURBINE WARSHIPS IN THE GERMAN NAVY.

Vessel	Date of Trials	Length	Beam	Draft	Displacement	Horsepower	Speed
*S*125.....	1905	205′	23′–0″	7′–0″	440	7,000	26.6*
Lubeck.....	1905	341	43–0	16–5	3,200	14,000	23.0*
*G*137.....	1907	233	24–9	7–10	570	13,000	33.09*
Steltin.....	1907	360	43–6	16–0	3,400	15,000	25.8*
Dresden.....	1908	365	44–3	16–0	3,600	17,500	25.0
Ersatz Greif...	(*a*) (*b*)	....	....	....	4,300	20,000	25.5
Cruiser *F*.....	(*a*)	530	85–6	27–0	19,600	44,000	24.25

* Speed on trial. (*a*) Ordered, October, 1907. (*b*) Three ships.

CLYDE RUDDERS AND RUDDER POSTS.

Fig. 1 shows the general practice of sternpost and rudder for the usual paddle steamer of 210 feet between perpendiculars by 25 feet beam by 9 feet molded depth. It will be seen that the rudder frame, stock and arms form one complete forging. The sternpost is also a forging, and is scarfed to the bar keel two and one-half frame spaces from the after perpendicular. The rudder stock is 4½ inches diameter, bushed at deck and top of cants, a distance of 5 feet 4 inches above load waterline; above the stock at upper bush a brass cap is fitted when the portable tiller is not required.

A quadrant is fitted above the cant frames, of the three-legged type, with a radius of 3 feet 6 inches. The legs are set to angles of 40 degrees. The boss is 8½ inches over all by 4 inches deep; 2 inches of metal are thus left on each side of the stock. The legs are 3¼ inches wide by 2½ inches deep at boss, and 2½ inches wide by 2 inches deep at palms. The palms are 7 inches deep by ⅝ inch thick. At the boss of tiller two snugs are fitted, to take 1¼-inch screws; to the screws are fitted the steering chains. A plate, 7½ inches deep by ⅜ inch thick, with double angles 3½ by 3½ by 7/16 inches, is riveted to palms of quadrant legs. On upper side of plate are riveted four horns, to prevent the chain from jumping.

The rudder arms are part of the forging. They are 4 inches

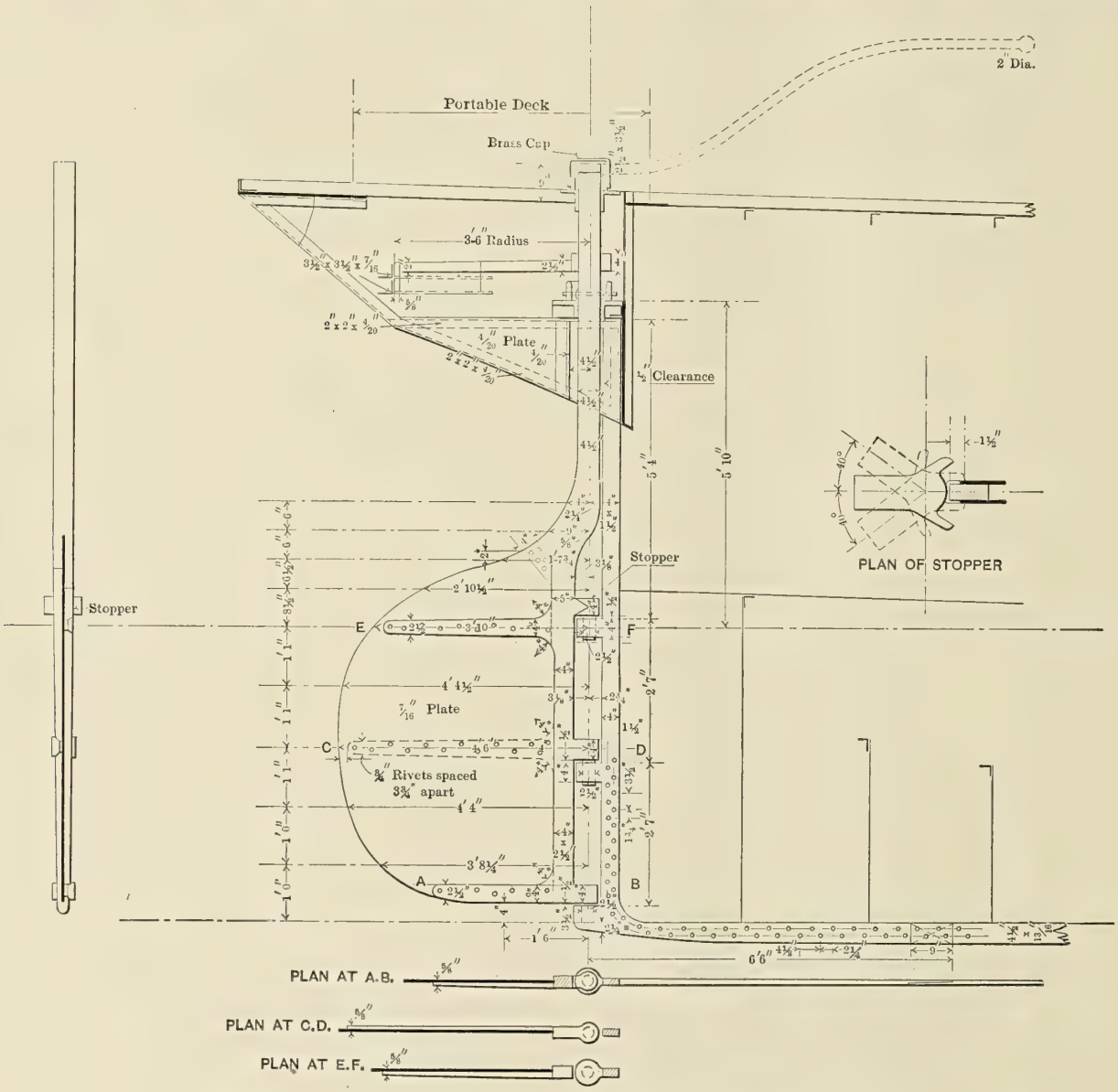


FIG. 1.—STERNPOST AND RUDDER FOR PADDLE STEAMER 210 FEET BY 25 FEET BY 9 FEET.

deep at fore side, 2½ inches deep at after side, and ⅝ inch thick at after end (see plan *C D*). The rivets through arms and rudder plate are ¾ inch diameter, spaced 3¾ inches apart. The rudder plate is of mild steel, 7/16 inch thick. The pintles are three in number, spaced 2 feet 7 inches apart; the diameters are 2½ inches. This leaves 1½ inches metal on each side of gudgeon. From underside of pintle bracket to top of gudgeon there is ½ inch clearance, and between gudgeon and rudder frame the clearance is ½ inch; between pintle bracket and sternpost the clearance is ½ inch. The lower pintle rests on a steel cap. The distance from center of pintles to rudder frame is 3⅛ inches, and from sternpost to center of pintles is 2¾ inches.

A detail sketch is shown of stopper, enabling the rudder to work through an arc of 80 degrees; the detail shows an arrangement which has found favor with ship owners. Another form of stopper in common use is made by fitting lugs to sternpost.

The sternframe is a forging, connected to bar keel by a scarf 9 inches long and 6 feet 6 inches from center of pintles. The gudgeons on it are 4 inches deep.

The top of sternpost is 5 feet 4 inches above load waterline, and is connected to 5/20-inch transom plate by double angles 2½ by 2½ by 5/20 inches. The rivets in vertical part of sternpost are ⅝ inch diameter, spaced 3½ inches apart (reeled). The rivets in sternpost in a fore-and-aft direction are ¾ inch diameter, spaced 4½ inches apart (reeled). The bar keel is 4½ by 13/16 inches.

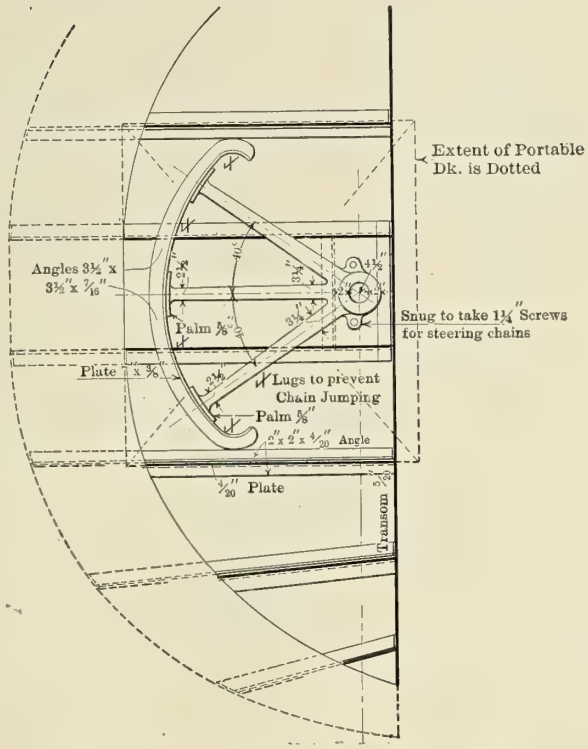
The rudder trunk is formed as follows:

Fore end, 5/20-inch transom plate.

After end, 4/20-inch plate, 4½ inches aft of center of rudder stock.

Sides, cant frames, 12 inches each side of center line.

Top, 6/20-inch plate, to take rudder bush.


Extent of Portable Dk. is Dotted
Angles 3½" x 3½" x 7/16"
Palm ⅝"
Plate ⅝ x ⅜
Lugs to prevent Chain Jumping
Palm ⅝"
2" x 2" x 4/20" Angle
4/20" Plate
Transom 5/20
Snug to take 1¼" Screws for steering chains

PLAN OF QUADRANT FOR 210-FOOT PADDLE STEAMER.

It will be noticed on plan that deck is portable above quadrant; the portable portion is indicated by dotted lines.

Fig 2 shows the sternpost and rudder of a single screw steam yacht 162 feet on the waterline. The rudder

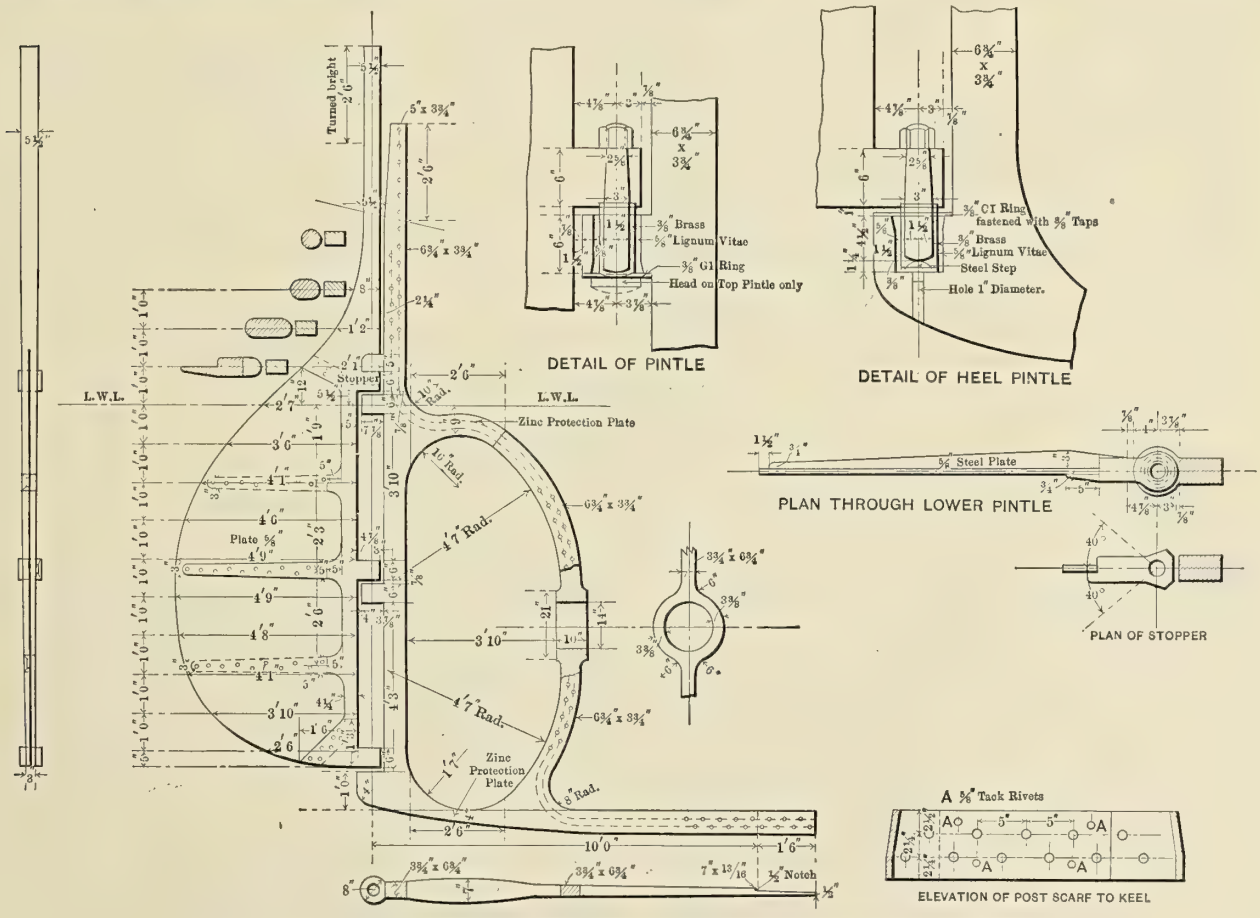


FIG. 2.—STERNPOST AND RUDDER OF SINGLE-SCREW STEAM YACHT 162 FEET ON THE WATERLINE.

frame and arms form one complete forging. The rudder stock is 5½ inches diameter, and this diameter is kept down to stopper, when a gradual taper commences, finishing at bottom pintle with a width of 3 inches. The rudder frame tapers in a fore-and-after direction from 5 inches to 4¼ inches. The arms are three in number, and are part of rudder frame. The upper arm is 21 inches below load waterline; the middle arm is 2 feet 9 inches below upper arm; the lower arm is 2 feet 6 inches below middle arm. The arms are 5 inches deep by 3 inches thick at fore end, tapering to 3 inches at after end by ¾ inch thick. A detailed sketch is given showing a plan through lower pintles. The rivets connecting arms to rudder plate are 1 inch diameter, spaced 5 inches apart. The rudder plate is ⅝ inch thick.

The three pintles are 3 inches diameter through gudgeons, tapering to 2⅝ inches. Each pintle is bushed with ⅜-inch brass and ⅝-inch lignum vitæ. ⅜-inch plate is fitted at bottom of upper and lower pintles, attached to gudgeons with ⅜-inch taps. A head is fitted on top pintle only (see detail). The bottom pintle rests on a steel cap as shown; the diameter and bushing are the same as for upper and lower pintles.

It will be noticed on detail that the lignum vitæ bush is slightly wedged. The pintle brackets and gudgeons are 6 inches deep. The clearances all around between rudder frame and sternpost are ⅞ inch. The distance from after side of post to center of pintle is 3⅞ inches, and from center of pintle to fore end of rudder frame 4⅞ inches. The clearance between bottom pintle bracket and lower gudgeon is 1 inch. The distance that bottom pintle is sunk into bottom gudgeon is 4½ inches, and 1¼ inches of steel cap is allowed for.

The rudder is arranged to work through an arc of 80 degrees before stopper is brought to bear hard against sternpost (see detail of stopper). From underside of bottom pintle to top of rudder stock the distance is 19 feet. The sternpost and propeller shaft post form one forging, 6¾ by 3¾ inches, scarfed to a bar keel 7 by 1 13/16 inches (as shown in detail).

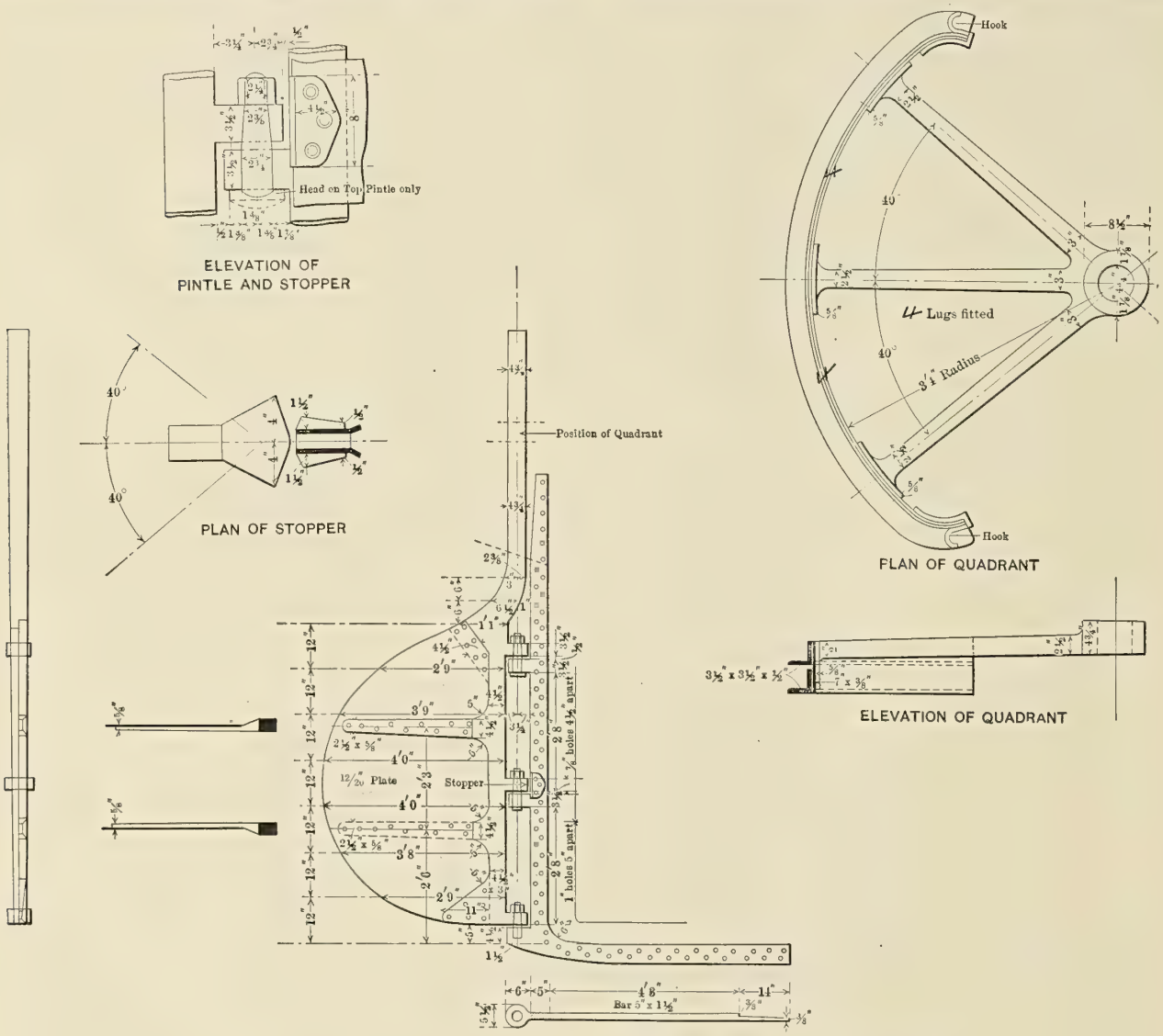


FIG. 3.—STERNPOST AND RUDDER FOR PADDLE STEAMER 165 FEET LONG BETWEEN PERPENDICULARS.

The bottom of the aperture is flattened to 7 by 4 inches. The aperture in sternpost is 3 feet 10 inches wide by 9 feet 9 inches deep. The boss is 10 inches wide by 3⅜ inches thick, and 21 inches over all. The 14-inch diameter is a rough bore only. The rivets in propeller post and part of sternpost are 1 inch diameter, spaced 5 inches apart, and the distance between centers in a fore-and-aft direction is 2¼ inches. The sternpost is carried up 30 inches above line of counter. Zinc protection plates are fitted for a length of 30 inches at top and bottom of aperture.

Fig. 3 shows the sternpost and rudder for a small paddle steamer 165 feet long between perpendiculars. The rudder stock, frame and arms form one complete forging. The rudder stock is 4¾ inches diameter; this diameter is kept to

upper pintle, whence the thickness is gradually reduced to 3 inches at lower pintle. The arms are 27 inches apart, and taper in depth from 4½ inches at rudder frame to 2½ inches at end of rudder. The fastenings through arms and rudder plate are ⅞-inch diameter rivets, spaced 5 diameters apart (reeled). The three pintles are 2¾ inches diameter.

The spacing of gudgeons is 2 feet 8 inches apart; the depth of gudgeons and pintle brackets is 3½ inches. The clearance all around between rudder and sternpost is ½ inch, and the space between rudder and sternpost is 6½ inches. The centers of pintles are 3¼ inches from sternpost and 3¼ inches from end of rudder frame.

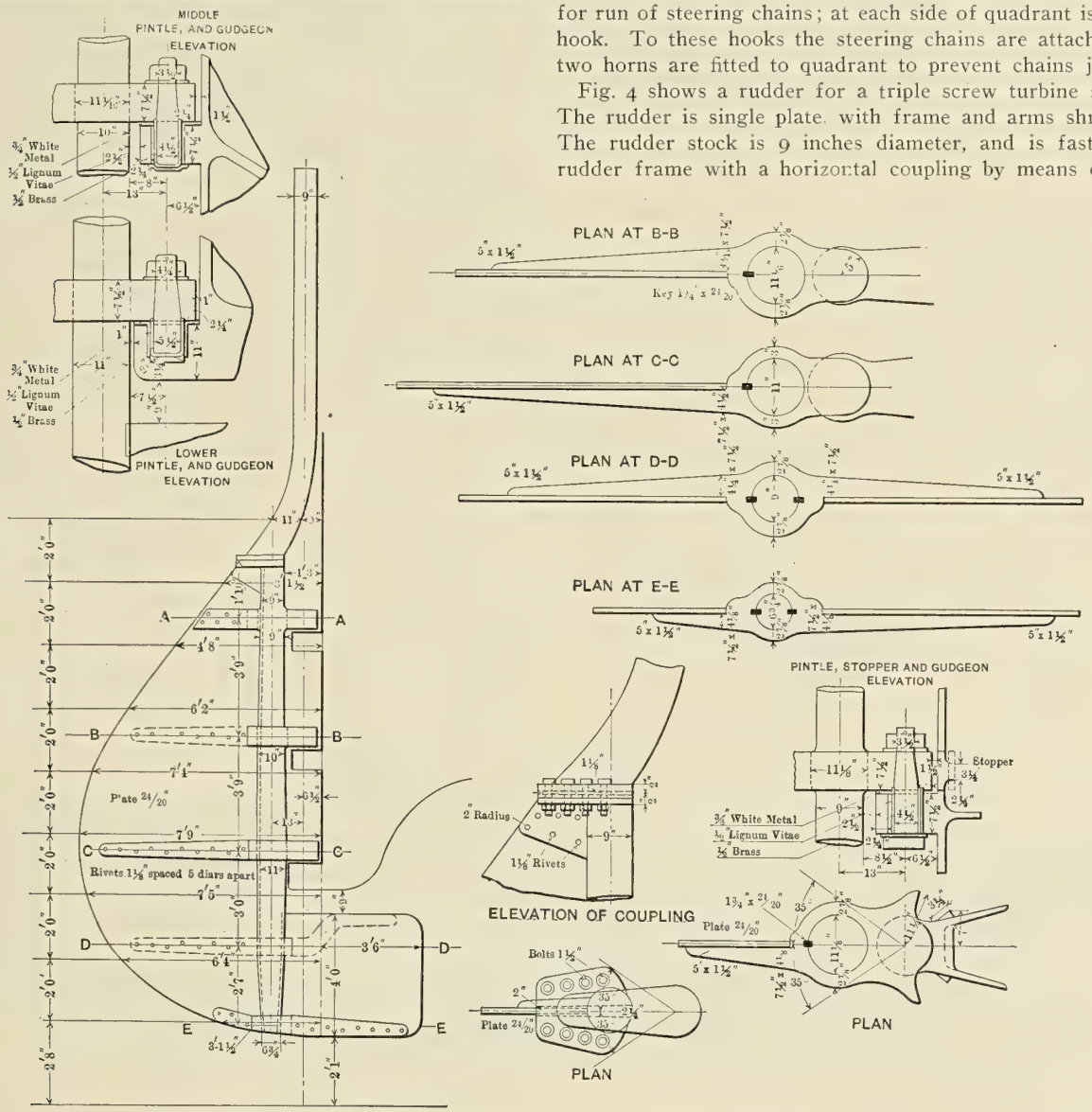


FIG. 4.—RUDDER AND DETAILS FOR A TRIPLE-SCREW TURBINE STEAMER.

A detail sketch of pintle, gudgeon and stopper is shown. This form of stopper shows a lug riveted to sternpost with three rivets. The lug is 8 inches long, 4½ inches broad, 1½ inches thick at after end and ½ inch at fore end. The stopper is on lower pintle, and is arranged to allow rudder to work through an angle of 80 degrees. The rudder plate is 12/20 inch thick.

The sternpost is a forging 10 feet 6 inches over all, of 5 by 1½-inch section. The rivets in upper part are ⅞ inch diameter, spaced 4⅜ inches apart; in lower portion the rivets are 1 inch diameter, spaced 5 inches apart. This spacing is continued to bar keel. The length of sternpost in a fore-and-aft direction is 5 feet 3 inches.

The quadrant is of the three-legged type, with legs at an angle of 40 degrees. The radius of quadrant is 3 feet 4 inches. The legs at boss are all 3 inches wide by 2½ inches thick, and 2½ inches wide by 2 inches thick at palm. The palms are 7 inches deep by ⅝ inch thick. The boss is 8½ inches diameter by 4¾ inches thick. After deducting 4¾ inches for diameter of rudder stock this leaves 1⅞ inches on each side of stock.

To palms of quadrant legs is riveted a 7 by ⅜-inch plate; and to the plate are riveted two 3½ by 3½ by ½-inch angles for run of steering chains; at each side of quadrant is fitted a hook. To these hooks the steering chains are attached, and two horns are fitted to quadrant to prevent chains jumping.

Fig. 4 shows a rudder for a triple screw turbine steamer. The rudder is single plate with frame and arms shrunk on. The rudder stock is 9 inches diameter, and is fastened to rudder frame with a horizontal coupling by means of eight 1½-inch turned bolts, and a tapered key, 2¼ and 2 inches by 1⅛ inches thick, is driven up the center of the coupling. The coupling is fitted above the waterline, so that the rudder can be unshipped without in any way interfering with the steering gear. The diameter of rudder frame at coupling is 9 inches; at middle pintle 10 inches; at bottom pintle 11 inches, and at bottom of rudder 6¾ inches.

The arms are shrunk on and secured against rotation by keys. A longitudinal groove for the plate serves as a keyway for the arms. The five arms are spaced from 3 feet 9 inches to 2 feet 7 inches apart. They are 7½ inches deep at rudder frame, and vary in thickness from 4⅛ to 4½ inches. At the end of the rudder they are 5 inches deep and 1½ inches thick. The thickness of arm bosses varies from 2⅞ to 3 inches.

The upper and lower pintles are 4½ inches diameter

through gudgeons; then the diameter is reduced to 3½ inches. The diameter of bottom pintle is 5½ inches, and tapers to 4¼ inches. This pintle rests on a steel button. A split pin is fitted above nut of each pintle. A head is fitted on upper pintle only. The pintles are bushed as follows: ½ inch brass, ½ inch lignum vitæ, with ¾ inch white metal tapped on to top of gudgeons. The rivets for connecting arms to rudder plate are 1⅛ inches diameter, spaced 5⅝ inches apart. The thickness of rudder plate is 24/20 inch.

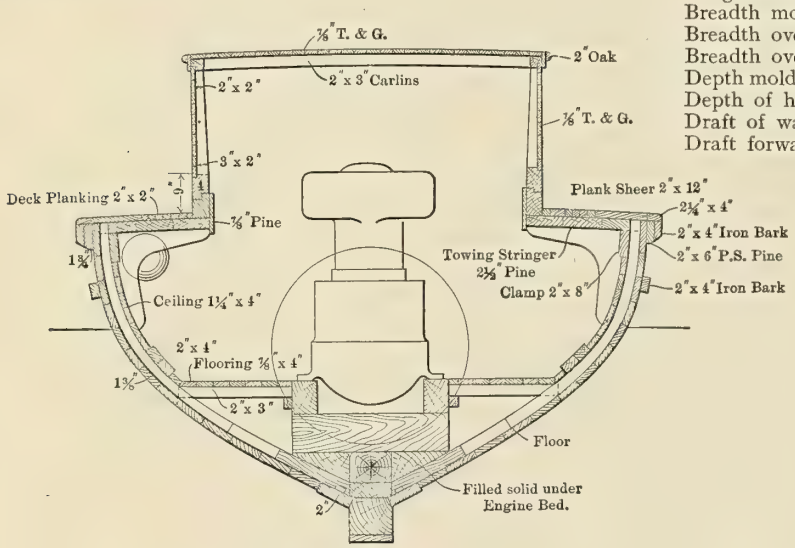
The distance from sternpost to center of pintles is 6½ inches, and from center of pintle to center of rudder frame is 13 inches. The clearance between rudder frame and sternpost at bottom pintle is 1 inch; at upper and lower pintles the clearance between sternpost and arms is 1½ inches; between gudgeon and rudder frame 2½ inches. The stopper is arranged to allow the rudder to work through an arc of 70 degrees.

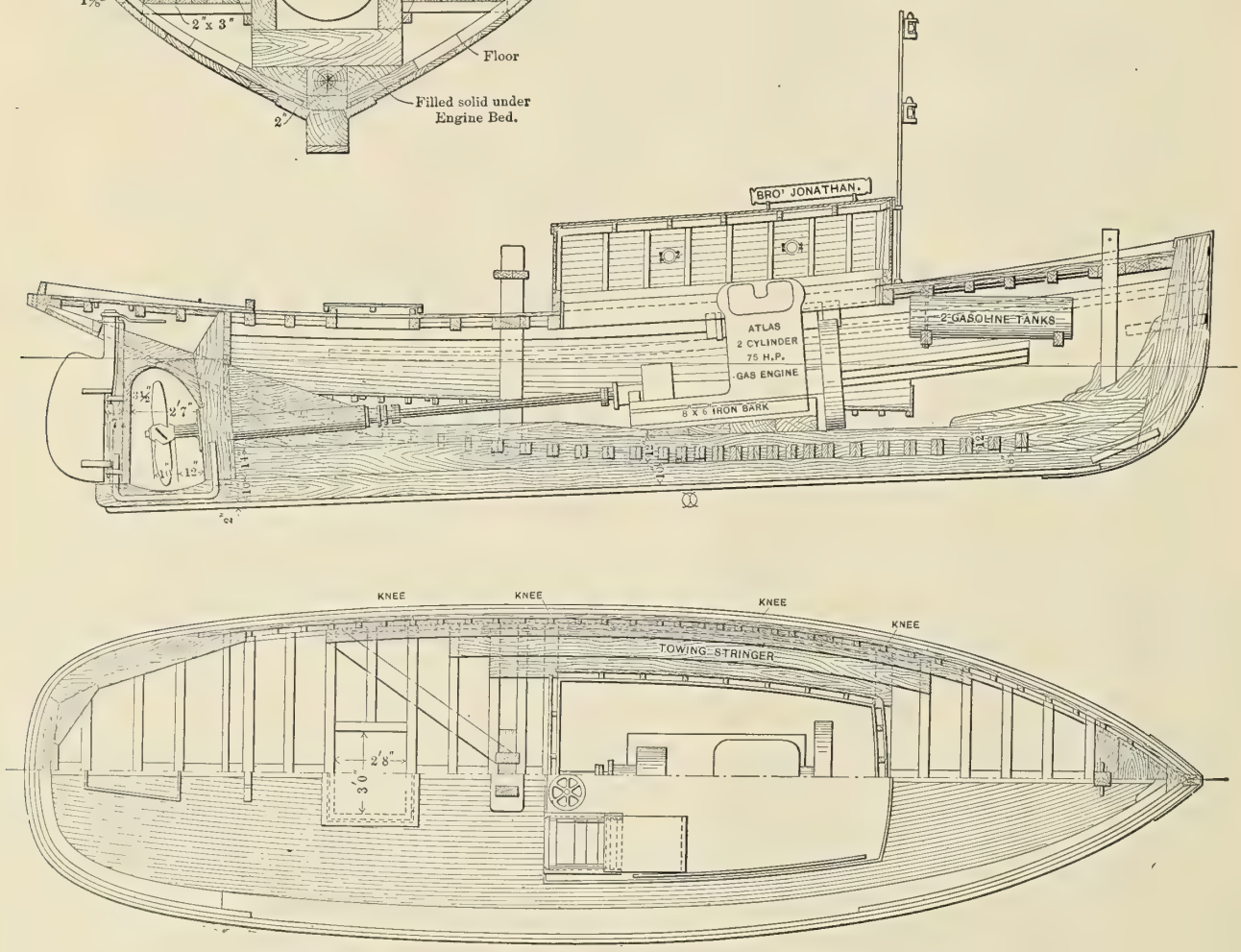
The Gasoline Towboat Brother Jonathan.

The use of the internal combustion engine for marine motive power in towboats is growing rapidly. One of the latest examples is the new gas tug *Brother Jonathan,* designed by D. W. & R. Z. Dickie, San Francisco, for James Wilder, and used in towing work in San Francisco bay. This type of boat has been developed by Mr. Wilder after many years of experience in operating boats of different types, and has proved very successful for towing barges, schooners and general small towing work.

Most of the towing is done on a headline, so the main towing bit is placed well forward to facilitate handling. The hull is of strong, plain construction, well fastened throughout with galvanized iron, the general dimensions being as follows:

	Feet.	Inches.
Length from forward side of stem to after side of stern post	37	10
Length over all	42	10
Breadth molded	11	3½
Breadth over planking	11	7
Breadth over guards	12	3
Depth molded, top of keel to top of beams at side	5	4
Depth of hold	4	6
Draft of water, aft, complete	5	6
Draft forward, 3 feet 11 inches; mean	4	8½





MIDSHIP SECTION, INBOARD PROFILE AND PLAN OF GASOLINE TOWBOAT BROTHER JONATHAN.

SIDE VIEW OF THE BROTHER JONATHAN.

The hull is short, to enable the boat to turn easily between the wharves. The steering, reversing and governing are all controlled by auto gear, the boat being easily handled by one man.

The lumber used for planking, decking, ceiling and housing is No. 1 Puget Sound pine. The stem, stern post, rudder, rudder post, cleats and bits are of oak. Five natural knees are fitted on each side, as shown on the plan, through fastened through deck beams and deck and frames and outside planking, which makes the boat very solid. The crown of deck beams is 4 inches in the width of 11 feet 7 inches.

The engine is of the two-cylinder heavy duty marine type, built by the Atlas Gas Engine Company, San Francisco, and develops 75 brake horsepower at 320 revolutions per minute, running on "No. 2" distillate. When running light the engine makes 320 revolutions per minute, turning a four-bladed elliptical propeller of 46 inches diameter, 40 inches pitch and 33 percent surface ratio, driving the boat at a speed of 10 miles per hour. The boat is too short for speed, the photographs showing excessive bow wave when running at full speed.

Type of engine (75 horsepower) ..	Steam	Gas
Fuel used.....	Crude oil	Distillate
Price of fuel.....	$1.25 per bbl.	7c per gal.
Cost of machinery per horsepower	$100 (£20½)	$50 (£10¼)
Weight per horsepower	450 lbs.	100 lbs.
Fuel per horsepower-hour.....	1.3 lbs.	⅛ U. S. gal.
Cost of same.....	42½c (21*d*)	63.8c (31*d*)
First cost: Hull.....	$2,900 (£600)	$1,800 (£370)
Machinery.....	$7,500 (£1,540)	$3,750 (£770)
Total.....	$10,400(£2,140)	$5,550 (£1,140)
Labor, cost per hour.....	$1.20 (4/11)	$0.60 (2/5½)
Total cost of operation per hour..	$1.625 (6/8)	$1.238 (5/1)

THE BROTHER JONATHAN AT FULL SPEED.

The first cost and cost of operation of a gas and steam tug of the same power is approximately shown by the table.

It will be seen from the table that the first cost and cost of operation of a gas tug of small power is much less than that of a steam tug, even though the fuel cost in the former is higher, the cost of labor accounting for the difference. The steam tug requires a hull 50 feet long, 14 feet beam and 6 feet 6 inches draft.

MARINE ENGINE DESIGN.

BY EDWARD M. BRAGG, S. B.

Calculations for the Connecting Rod.

Let $l = 2.25 \times \text{stroke} = 2.25 \times 42'' = 94.5''$

$k = 1.026$;

$P = 77{,}700 \times 1.026 = 79{,}800$ pounds;

$N = 12$;

$C = 70{,}000$ pounds;

$$f = \frac{70{,}000}{12} = 5{,}825 \text{ pounds.}$$

$$F = \frac{2 \times 79{,}800}{3.142 \times 5{,}825} = 8.73.$$

Formula (27):

$$D^2 = \sqrt{\frac{1.08 \times 8.73 \times 70{,}000 \times (94.5)^2}{10{,}000{,}000}} + 76.2 + 8.73 = 34.53.$$

$D = 5.88''$, (Use $5\frac{7}{8}$ inches), diameter at middle of rod.

Diameter at top of rod $= 5.875 \times 0.9 = 5.28$, (Use $5\frac{1}{4}$ inches);

Diameter at bottom of rod $= 5.875 \times 1.1 = 6.45$, (Use $6\frac{1}{2}$ inches).

Load on bolts of crosshead pin caps $= \frac{79{,}800}{4} = 19{,}950$ pounds; this will require 2.5-inch bolts (see Table IV.).

$$T = 6.25'' + 2.5'' + 0.5'' = 9.25''.$$

Heads of bolts will be $1.5 \times 2.5'' = 3.75''$ diameter.

$9.25'' - 3.75'' = 5.5''$. The thickness Z of the fork can be $5\frac{3}{8}$ inches and have the heads of the bolts clear the fork by $\frac{1}{16}$ inch. The length of the crosshead pin = 6.25 inches. Let the overhang of the box be $\frac{3}{4}$ inch on each side; $K = 6.25'' - 1.5'' = 4.75''$. We will make the caps of wrought steel.

Formula (29): $$S = \sqrt{\frac{79{,}800 \times 9.25}{2 \times 4.75 \times 7{,}500}} = 3.22'' \text{ (Use } 3\tfrac{1}{4}''\text{)}.$$

$F = 9.5'' + .25'' = 9.75''$;

$W = 9.75'' + 1.5'' = 11.25''$;

$V = 4.375'' + 4'' + 4'' = 12.375''$;

$Q = 2.5'' + .5'' = 3''$;

$N = 1.5''$.

The radius of the inside of the fork is $\frac{W}{2} = 5.625$ inches.

Draw lines parallel to XX, Figure 31, spaced 1 inch apart.

Formula (30): $$h^2 = \frac{3 \times 79{,}800 \times 1}{5.375 \times 5{,}000} = 8.92; \quad h = 2.985 \text{ inches.}$$

$h^2_2 = 2 \times 8.92 = 17.84$; $h_2 = 4.23$ inches.

$h^2_3 = 3 \times 8.92 = 26.76$; $h_3 = 5.18$ inches.

$h^2_4 = 4 \times 8.92 = 35.68$; $h_4 = 5.97$ inches.

$h^2_5 = 5 \times 8.92 = 44.60$; $h_5 = 6.68$ inches.

The dimensions of the collar nuts can be taken from Table V.; total thickness of liners for crosshead end to be 2 inches, as follows:

1 cast iron liner $1\frac{5}{8}$ inches thick,

1 brass liner $\frac{1}{4}$ inch thick.

1 tin liner	$1/16$ inch thick.
1 tin liner	$1/32$ inch thick.
2 tin liners	$1/64$ inch thick.

Each bolt of the crank pin box must carry $\frac{79,800}{2} = 39,900$ pounds.

From Table IV., the diameter should be $3\frac{1}{4}$ inches.

Formula 37 (see below under crank shafts) gives the size of the crank shaft.

$$D = (0.038 \times 23.5 + 0.009 \times 41 + 0.002 \times 64. + 0.0165 \times 42.) \times \sqrt[3]{185}$$
$$= 11.9'', \text{ (Use 12 inches).}$$

Make the diameter of the crank pin $12\frac{1}{2}$ inches.

Formula (31): $L = 1.6 \times \frac{21,000 \times 1,000}{850} = 39,500$ pounds.

From Table VII., allowable pressure = 225 pounds. $\frac{39,500}{225} = 175.5$ square inches. Length C of crank pin $= \frac{175.5}{12.5} = 14$ inches. Let the box overhang 2 inches on each side; then $L = 10$ inches. $U = 12.5'' + 3.25'' + 0.75'' = 16.5''$. Make the cap of cast steel.

Formula (32): $R = \sqrt{\frac{79,800 \times 16.5}{10 \times 5,000}} = 5.12''$, (Use $5\frac{1}{4}$ inches).

Let $M = 2''$, $O = 0.625''$, $P = 3.75''$. Total thickness of liners for crank-pin end to be $3\frac{1}{2}$ inches, as follows:

1 cast iron liner	$2\frac{3}{4}$ inches thick.
1 brass liner	$9/16$ inch thick.
2 tin liners	$1/16$ inch thick.
1 tin liner	$1/32$ inch thick.
2 tin liners	$1/64$ inch thick.

SHAFTING.

Crank Shaft.—The stress upon the crank shaft can be figured by means of the formula $f = \frac{My}{I}$,

where M = the maximum equivalent twisting moment;
y = radius of shaft;
and I = polar moment of inertia.

or $f = \frac{TD}{\frac{4\pi D^4}{64}} = \frac{16T}{\pi D^3}$.

For solid shafts, $D = \sqrt[3]{\frac{T \times 5.1}{f}}$. (33)

For hollow shafts, $f = \frac{T \times D \times 5.1}{D^4 - d^4}$. (34)

The maximum twisting moment can be obtained from the mean twisting moment by means of a factor derived from analyses of engine trials. The mean twisting moment

$$= \frac{I.H.P. \times 33,000}{2\pi n}, \quad (35)$$

where $I.H.P.$ = total indicated horsepower of engine,
and n = revolutions of engine per minute.

The maximum can be obtained from this by multiplying by a factor C, which has the following values:

Single-cylinder engine,	$C = 2.$
Two-cylinder engine,	$C = 1.5$
Three-cylinder engine,	$C = 1.33$
Four-cylinder engine,	$C = 1.25$

The crank shaft is subjected to bending as well as twisting, and the equivalent twisting moment is usually found by use of the formula:

$$T_1 = M + \sqrt{M^2 + T^2}, \quad (36)$$

where M = the bending moment at the section where
T = the maximum twisting moment.

The load upon the crank shaft is intermittent in character, varying from a maximum to a minimum. The factor of safety to be used, then, is 8 if the twisting moment is that given by formula (36). If the bending moment is neglected, and only the twisting moment is used in figuring the shaft, the factor of safety should be increased to 10.

In order to get a rating it is necessary that not only a ship's hull but also certain parts of its machinery shall conform to the rules of the registering society. The crank shaft is one of the parts of machinery whose size is specified, so that its diameter will be determined by some of the following rules, rather than by the preceding formulæ.

Lloyd's Rules (1906-1907).—Stroke greater than ½ low-pressure diameter and less than low-pressure diameter.

Three Cranks at Equal Angles.—Ratio of low-pressure area to high-pressure area not over 9:

Diameter of line shaft

$$= (0.038A + 0.009B + 0.002D + 0.0165S) \times \sqrt[3]{P}. \quad (37)$$

Four Cranks at Equal Angles.—Ratio of low-pressure area to high-pressure area not over 12:

Diameter of line shaft

$$= (0.033A + 0.01B + 0.004C + 0.0013D + 0.0155S) \times \sqrt[3]{P}. \quad (38)$$

Diameter of crank shaft = 21/20 diameter of line shaft.

Diameter of thrust shaft between collars = 21/20 diameter of line shaft.

Diameter of screw shaft

$$= \left(0.63 + \frac{0.03P}{T}\right) \times T,$$

where P = diameter of propeller, and T = diameter of line shaft. In no case must the diameter of screw shaft be less than 1.07 T.

In the above,

A = diameter of high-pressure cylinder in inches;
B = diameter of first medium-pressure cylinder in inches;
C = diameter of second medium-pressure cylinder in inches;
D = diameter of low-pressure cylinder in inches;
S = stroke of engine in inches;
P = gage boiler pressure.

Bureau Veritas Rules (1907) for shafting of double, triple and quadruple expansion engines, no overhung cranks:

$$d = \sqrt[3]{\frac{PL(n_1 D_1^2 + 0.1nD^2)}{C}}, \quad (39)$$

where d = diameter of after shaft bearing in inches;
n_1 = number of high-pressure cylinders;
D_1 = diameter of each high-pressure cylinder in inches;
n = number of low-pressure cylinders.
D = diameter of each low-pressure cylinder in inches.
L = length of stroke, in inches, common to all pistons.
P = gage boiler pressure in pounds per square inch.

For hollow shafts the diameter must be increased by 1 percent if the diameter of the hole is 0.4 of the outside diameter; by 2 percent if the diameter of the hole is 0.5 of the outside diameter; by 5 percent if the diameter of the hole is 0.6 of the outside diameter, and by 10 percent if the diameter of the hole is 0.7 of the outside diameter.

If the sequence of cylinders and angles of the cranks in quadruples are so chosen as to reduce the maximum torsional

Type of Engine.	No. of Cylinders.	No. of Cranks.	Angle Between Cranks.	Value of Factor C.
Compound.....	2	2	90°	3,400
Compound.....	4	2	90°	3,500
Compound.....	6	3	120°	3,800
Compound.....	2	1	....	2,100*
Compound.....	3	3	120°	3,600
Triple.....	3	3	120°	3,900
Triple.....	3	2	90°	3,000
Triple.....	4	2	90°	3,300
Quadruple.....	4	2	90°	3,100†
Quadruple.....	4	4	90°	4,000†

* Cutoff at 0.8 in high-pressure cylinder.
† Subject to approval.

moment the above values may be increased, but in no case shall they exceed 4,100. Each particular case must be submitted for approval.

The diameter of propeller shaft must be $(1.7 \frac{D}{d} - 15)$ percent in excess of the diameter of the crank shaft; where D = diameter of propeller in inches, and d the diameter of crank shaft in inches. The diameter of thrust shaft at the bottom of collars, both between and immediately beyond these latter, to be equal to that of the crank shaft, and tapered off at each end to the smaller diameter of the body of the shaft. For tunnel shafts, a reduction of 6 percent on the diameter of the crank shaft will be allowed.

Turbine Shafting.—In turbine engines, where *I. H. P.* is the estimated power transmitted by each shaft,

$$d = \sqrt[3]{\frac{70 \times I.H.P.}{R}}; \quad d_1 = d + \frac{D}{160}; \quad d_2 = 1.05d, \qquad (40)$$

where d = diameter of tunnel shafting in inches;
d_1 = diameter of propeller shaft in inches;
d_2 = diameter of rotor shaft in inches at smaller part;
D = diameter of propeller in inches;
and R = number of revolutions per minute.

American Bureau of Shipping Rules for Shafting.

$$S = \sqrt[3]{\frac{PD^2L}{K\left[\frac{D^2}{d^2} + 2.4\right]}}, \qquad (41)$$

where S = diameter of crank shaft in inches;
P = gage pressure at boiler in pounds per square inch;
D = diameter of low-pressure cylinder in inches (if there is more than one low-pressure cylinder, then for D^2 use the sum of the squares of the diameters);
L = length of stroke in inches;
K = factor from following table:

Type of Engine.	No. of Cranks.	Angle Between Cranks.	Value of Factor K.
Compound.....	2	90°	1,450
Compound.....	2	180° or 0°	1,200
Compound (3 cylinders)...	3	120°	1,500
Triple.....	3	120°	1,700
Triple (4 cylinders).....	4	90°	1,600
Quadruple.....	4	90°	1,800
Quadruple (5 cylinders)....	5	72°	2,100

In four-crank engines with unequal spacing of cranks, K must be decreased as follows:

If smallest angle lies between 75 degrees and 85 degrees, decrease K by 100; if smallest angle lies between 65 degrees and 75 degrees, decrease K by 200; if smallest angle lies between 55 degrees and 65 degrees, decrease K by 300.

For hollow shafting see under equation 39. The diameter of the propeller shaft is to be 6 percent in excess of the diameter of the crank shaft. The diameter of the tunnel shafting may be 5 percent less than that of the crank shaft.

Crank shafts are generally made in sections, one section for each cylinder. The sections are flanged and joined together by coupling bolts. Shafts may be divided into two classes, forged and built-up, and still further divided into solid shafts and hollow shafts. In the merchant marine, shafts less than 12 inches diameter are usually forged, as shown in Fig. 32, except that they are seldom hollow; above that diameter they are usually built up, as shown by Fig. 33. Hollow shafts are seldom used in merchant ships, but are used almost altogether in vessels of the navy.

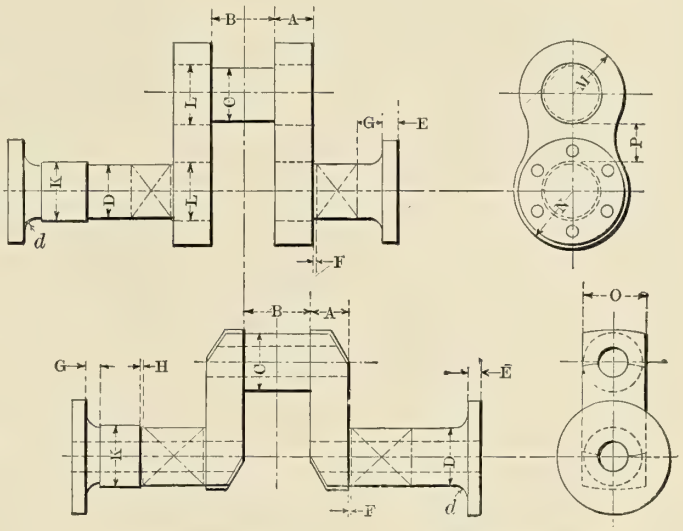


FIG. 32.—(BOTTOM). FIG. 33.—(TOP).

The proportions of the different parts should be as follows:

Diameter = D (see above).
Thickness of webs = A = 0.6D to 0.7D.
Length of crank pin = B = length of crank pin box + $\frac{1}{8}$ inch to $\frac{3}{8}$ inch.
Diameter of crank pin = C = D to D + 1 inch.
Thickness of couplings = E = 0.22D to 0.25D, naval engines.
= 0.25D to 0.28D, merchant engines.
Clearance between webs and main bearings = F = $\frac{1}{4}$ inch to $\frac{3}{8}$ inch.
Clearance between couplings and main bearings = G = 2 inches to 3 inches.
Clearance between eccentric pads and main bearings = H = $\frac{1}{4}$ inch to $\frac{1}{2}$ inch (minimum).
Radius of coupling fillet = d = 1 inch to 2 inches.
Diameter of eccentric pads = K = D + $\frac{1}{2}$ inch to D + $\frac{3}{4}$ inch.
Diameter of journal and pin in crank webs = L = C + $\frac{1}{2}$ inch to C + 1 inch.
Radius of crank web at pin = M = 0.875L.
Radius of crank web at shaft = N = 0.925L.
Breadth of web in solid shafts = O = 1.05D to 1.1D.
Metal between shaft and pin = P = 0.45L (at least).

When the three sections of shafting for a triple engine are to be interchangeable, all of the crank webs should be of the same thickness, otherwise the forward webs can be made thinner than the after webs, having a thickness of 0.6 D at the forward end and 0.7 D at the after end. The bearings can be of different lengths, but sufficient clear space must be

allowed on all sections of the shaft to accommodate the longest bearing, if the sections are to be interchangeable.

The diameter of the coupling flange must be determined by the size of the coupling bolts used. The pitch circle of the bolts must be large enough to permit the nuts on the ends of the bolts to turn, without having to cut away anything but the fillet of the flange. The correct number of bolts and diameter of pitch circle can be determined only by trial. In the first assumption, the radius of pitch circle can be taken as 0.75 of the diameter of the shaft, and in the largest sizes of shafts one bolt is allowed for about each 2 inches of diameter of shaft. In order that the shearing strength of the bolts may equal the torsional strength of the shaft, the diameter of the coupling bolt should be, for solid shafts,

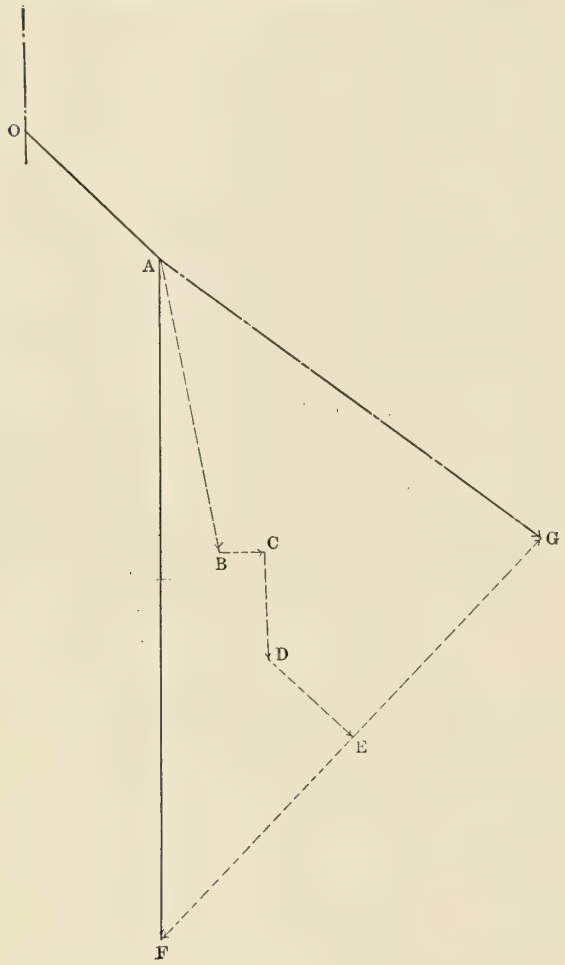


FIG. 34.

$$R = \frac{D}{2} \times \sqrt{\frac{D}{n \times r}}, \qquad (42)$$

and for hollow shafts:

$$R = \tfrac{1}{2}\sqrt{\frac{D^4 - d^4}{D \times n \times r}}, \qquad (43)$$

where R = diameter of coupling bolt at shearing section;
D = outer diameter of crank shaft;
d = inner diameter of hollow shaft;
n = number of coupling bolts;
and r = radius of pitch circle of coupling bolts.

The diameter of the coupling flange should be $2(r + R)$.

The coupling bolts are usually tapered from end to end at the rate of 1 inch per foot. Where the nut is attached at the smaller end the diameter of the bolt can be reduced from ¼ inch to ½ inch, as the nut serves merely to keep the bolt in place.

BEARINGS.

The diameter of the crank shaft in the main bearings is that given by the formula for crank shafts. The length of the bearings is made such that the bearing pressure shall not exceed that given in Table VII. The loads upon the bearings increase as we go towards the propeller, and are not all of the same character. In some cases the bearing pressure is well distributed over the entire circumference of the bearing; in other cases it tends to act upon a small portion only. As the important thing about a bearing is that it shall be sufficiently large to be kept cool, it can be seen that a larger unit bearing pressure can be allowed where the load is well distributed than where it is concentrated upon one part only.

When a piece of shafting, such as a line shaft, for instance, is transmitting power, there is no load upon the supports except that due to the weight of the shafting. If, however, a crank is introduced into the shaft, and a bearing placed on either side of the crank, these bearings will be subjected to loads in addition to those from weight alone. The force which one web delivers through the crank pin to the other web will be felt upon the bearing adjacent to the latter web, while the reaction which the first web experiences will be felt upon the bearing nearest that web. These loads upon the bearings must be equal in amount and opposite in direction, in order that the shaft which was in equilibrium before the crank was introduced may still be in equilibrium.

The turning force acting at the crank pin will increase as we go towards the propeller, so that this component of the bearing pressure will become more predominant. The components of the bearing pressure are shown in Fig. 34. OA is the position of the crank after it has turned through 135 degrees. AB is the force acting through the connecting rod, and is composed of the weight of the reciprocating parts and the unbalanced steam pressure on the piston. BC is the cross-throw of the connecting rod. CD is the resultant of the weight of the shaft and the inertia of the reciprocating parts. DE is the centrifugal force of the unbalanced parts of the crank webs and crank pin. EF is the force acting upon the after web, due to the turning force from the forward cylinders. EG is the reaction upon the forward web, due to the above force. AF is the resultant force acting upon the after bearing, and AG is the resultant for the forward bearing.

It will be seen that, if the turning force were left out, as is the case in the bearings of the first cylinder, the two bearings would have the same load upon them, and acting in the same direction. The effect of the turning force, however, is to cause the resultant loads to differ quite widely in amount and in direction. As the turning force changes its direction of action through 360 degrees during a revolution, its effect upon the resultant, as it becomes more predominant in the bearings nearer the propeller, is to cause the load to be distributed more uniformly around the bearing. In the case of the forward bearing of each pair of bearings, since the reaction of the turning force is, in general, acting in a direction opposite to that of the other forces, the resultant tends to act upon the sides of the bearings.

The mean load upon the bearings can be found approximately by means of the formula:

$$L = \frac{21{,}000}{PS}[HP_t + fHP_c], \qquad (44)$$

where PS = piston speed in feet per minute;
HP_t = the indicated horsepower developed by the cylinders forward of the cylinder whose bearings are in question;
HP_c = the indicated horsepower developed in the cylinder over the bearings;
and f = a factor whose value can be taken from the curves given in Figure 35.

The factor f allows for two components of the bearing load, one due to the power being developed in the cylinder over the bearings, and the other due to the centrifugal force of the rotating parts. If we had to deal with the first component only, f would be constant for all horsepowers, but since the centrifugal force of the rotating parts increases at a greater rate than the horsepower, the factor f will increase as the horsepower increases. When the centrifugal force of the rotary parts is balanced by weights upon the crank webs, the factor f will be taken as for zero *I. H. P.*

Sometimes the bearings between two cylinders are combined into a common bearing. In this case the load coming upon it

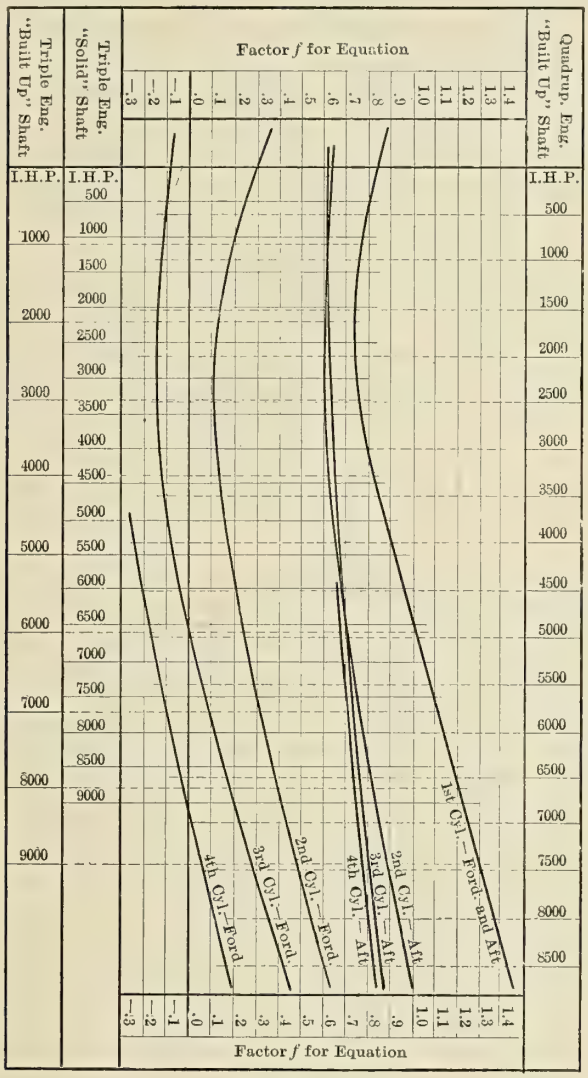


FIG. 35.

can be found approximately by calculating the load coming upon each bearing by means of formula 44, and multiplying the sum of the two loads by a factor taken from the following table:

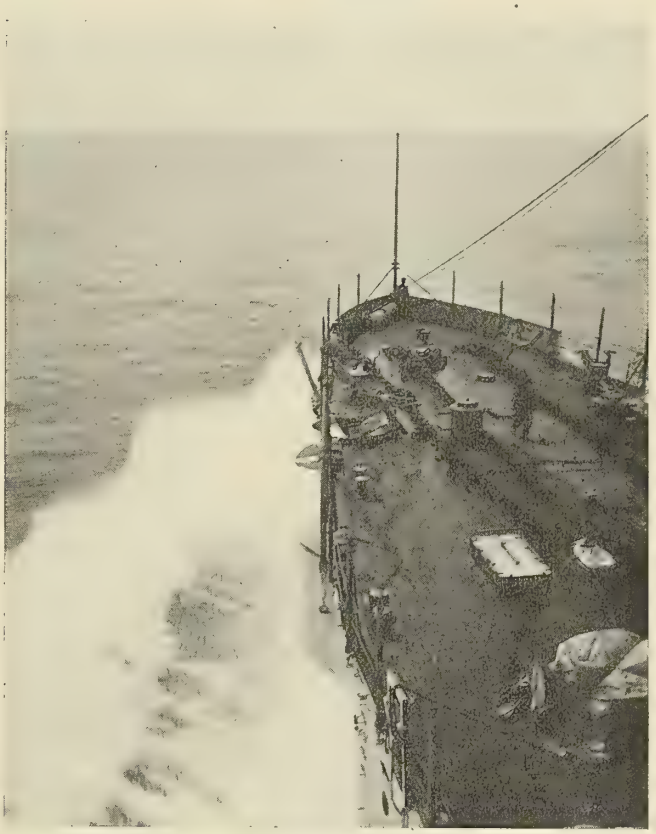
TABLE OF FACTORS FOR COMMON BEARINGS.

Between.	Type of Engine.	Angle Between Cranks.	Factor.
1st and 2d cylinders..	Triple; high leading....	120°	0.6
	Quadruple.....	180°	0.45
	Triple; low leading.....	240°	0.85
2d and 3d cylinders..	Quadruple.....	90°	0.47
	Triple; high leading....	120°	0.625
	Triple; low leading.....	240°	0.9
3d and 4th cylinders..	Quadruple.....	180°	0.875

When two cranks are at 180 degrees from one another, with a common bearing between them, the steam loads are opposed to one another, and balance, if equal powers are developed in the two cylinders. On the other hand, the loads from the turning force, being transmitted through the crank pins, are added to one another. This can be readily seen when it is remembered that the common bearing serves as the after bearing of one cylinder, and the forward bearing of the other cylinder. Since the cranks make an angle of 180 degrees, the two forces, which in the bearings for a single crank are opposed to one another, are in this case added to one another.

It will be found usually that in the case of triple-expansion engines the two bearings of the first cylinder and the forward bearing of the second cylinder can be made of the same size, also that the after bearing of the second cylinder and the forward bearing of the third cylinder can be made the same. The after bearing of the third cylinder will be the largest of all.

(To be Continued.)



BOW WAVE THROWN UP BY THE CRUISER MONTANA AT FULL SPEED.

Bow Wave of the Montana.

When the American armored cruiser *Montana* passed through her official trial trip last spring, and attained a mean speed for four hours of 22.26 knots with 27,489 indicated horsepower, a photograph was made of the bow wave when running full speed. This photograph was taken from the extreme port end of the flying bridge, and is here reproduced.

The *Montana* is a large armored cruiser, having a displacement of 14,500 tons at her normal draft of 25 feet. The length is 502 feet on the waterline, with an extreme beam of 72 feet 8 inches. The ship at this draft carries 900 tons of coal, but has stowage capacity for no less than 2,000 tons. The main battery includes four 10-inch rifles mounted in pairs in two turrets on the center line, forward and aft, and sixteen 6-inch rapid-fire guns, located twelve on the gun deck and four on the main deck. The secondary battery includes twenty-two 3-inch and twelve 3-pounder rapid-fire guns, six

machine guns and four 1-pounder automatic guns. There are four submerged torpedo tubes. The armor includes a belt 16 feet wide and 5 inches thick (3 inches at ends). The turrets and barbettes have, respectively, 9 and 7 inches of armor, maximum, while the guns in battery are protected by 5 inches. The armored deck varies from 1½ to 3 inches in thickness.

MARINE GASOLINE ENGINE DESIGN.

BY E. W. ROBERTS, M. E.

The object of this article is not to enter into all the lengthy details of gas-engine design, but to point out some special requirements for designing marine gasoline (petrol) engines according to modern ideas. With the exception of engines built to meet extreme racing requirements, lightness in pounds per horsepower, while desirable, is not the chief requisite. One should have, in a first class marine engine, strength, lack of vibration, stability, large wearing surfaces and certain lubrication as the major requirements to be met, and quite a number of minor requirements, that will be taken up during this discussion.

Strength, stability and lack of vibration practically go hand-in-hand. An engine lacking either one of these is a very poor engine for a boat, and will give endless trouble. While strength and lack of vibration are not obtained by the same methods, yet, to have good stability, an engine must be both strong and vibrationless.

There is an inclination among designers toward extremely light engines, especially in those of the "built-up" type, to sacrifice strength in order to get extreme light weight. This

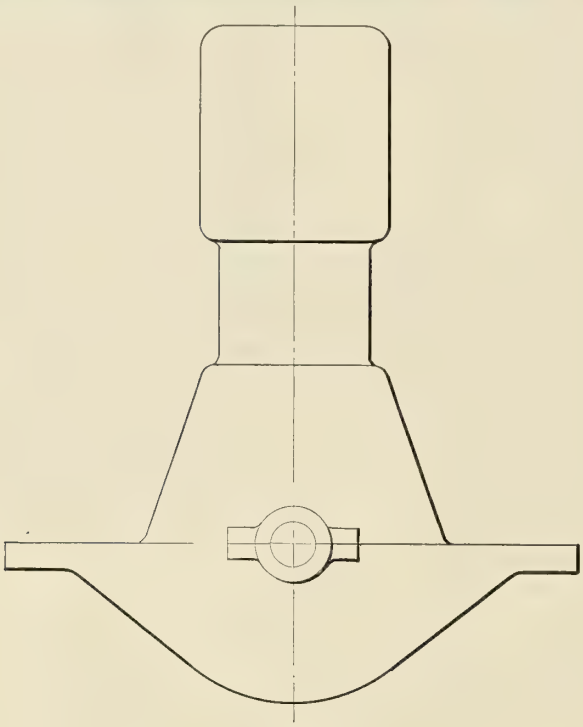


FIG. 1.

style of engine always reminds the writer of a little steam engine in one of his early boats, which was so flimsy that it had to be held when it got up to speed. The best style of frame for a four-cycle engine is what is known as an A-frame, and is of the general design shown in Fig. 1. This style gives strength without excessive weight, and has no chance to work loose from the motion of the engine.

Usually, for stability of operation, one of the chief requirements is a broad and long base for the engine to rest upon. On the earlier engines there was used a narrow frame, which made the engine inclined to rock on its foundation. Modern engines have a base extending outward, so that the foundation timbers may be carried forward of the fly-wheel, as shown

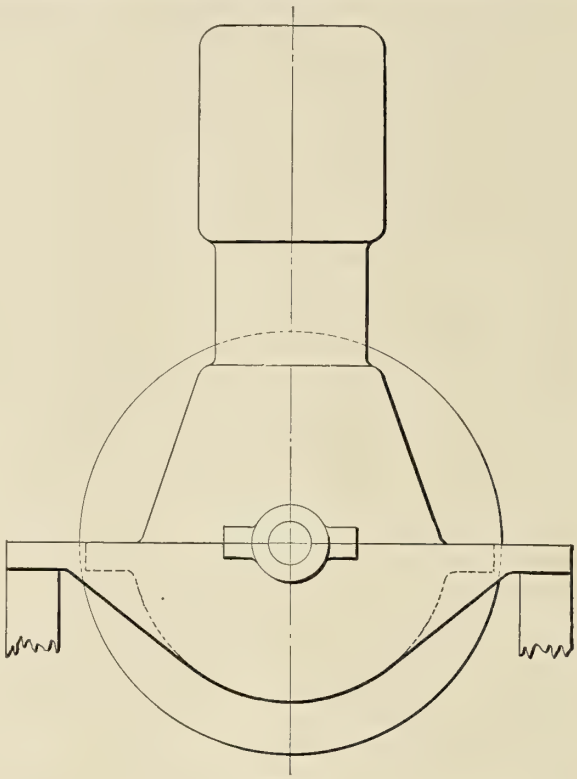


FIG. 2.

in Fig. 2 by the full lines. The old and the poorer way is shown by dotted lines.

Quite a number of marine engines are constructed with a flat bed-plate forming part of the base. In extending this to reach beyond the fly-wheel, it is likely to be made very heavy. This surplus weight can be avoided by using well-ribbed arms, instead of a plate. Quite a few boat-builders are inclined to think that this is detrimental to the working of the engine; but they will find, on inspection, that quite a number of automobile engines are built in this way. In large engines, with heavy fly-wheels, it is often necessary to make an extra broad arm near the fly-wheel, using, if the occasion requires, three ribs instead of two. Quite frequently it is of advantage to face the top of the foundation timbers with iron, in order that an arm may not be drawn too tightly into the foundation timber and, therefore, twist the base out of line.

The minimum of vibration is obtained in single-cylinder engines by careful counterbalancing. Generally this is done in a half-hearted way by guess, and no means are employed when the engine is being built for verifying the counterbalancing effect. Quite a number of builders are inclined to balance their single-cylinder engines in the fly-wheel. First-class results can be obtained only by counterbalancing with bob weights on the crankwebs. The best results, in a trunk piston engine, can be obtained by balancing with bob weights according to the following rule: Weigh carefully the piston with its piston pin and piston rings, and then weigh each end of the connecting-rod, by placing one end on the scale and balancing the other end on a knife edge, directly opposite the center of the bearing, and with the connecting-rod held horizontal. Then design a counter weight with a balancing effect equal to one-half the weight of the piston plus one-half the small end of the connecting-rod plus the whole weight of the large end of the connecting-rod and the crankpin, and with a lever arm equal to that of the crankpin. Add to this the leverage of the crankwebs, taken at the center of gravity. In other words,

balance half the reciprocating parts and all of the rotating parts.

Stated as a formula, it would be as follows:

Let P = weight of piston with pin and rings,
D = weight of piston end of rod,
C = weight of crank end of rod,
W = weight of web,
Q = weight of crankpin,
T = crank throw = one-half stroke,
S = radius to center of gravity of web,
R = radius to center of gravity of counterbalance, and
K = weight of counterbalance.

$$\text{Then } K = \frac{[\frac{1}{2}(P+D)+C+Q]\,T+W\,S}{R}$$

A bob weight may be cast from gray iron, with wrought iron straps cast on, as shown in Fig. 3, and the weight may be riveted to the crankwebs by means of these straps.

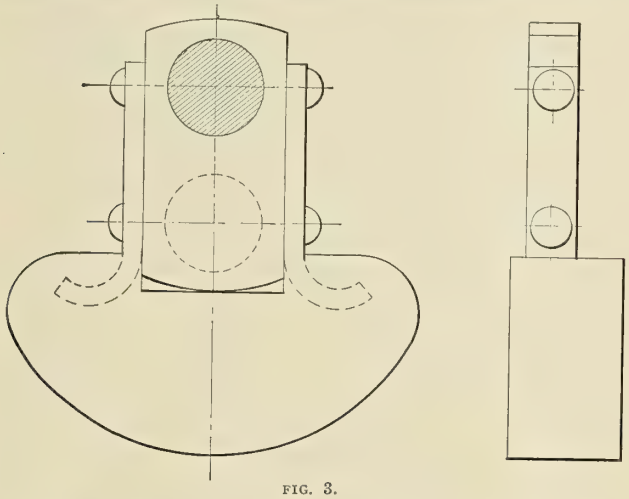


FIG. 3.

Finally, check up your balance weight, as shown in Fig. 4, laying the crankshaft on a pair of straight edges and hanging a weight W on the crankpin, as shown. As the crankpin and the crankwebs are used, the weight should equal $\frac{1}{2}(P+D)+C$. The weight may be added to by drilling with an under cut and filling with lead, and weight may be taken out by drilling. It takes but a moment to make this test, and it should be done with every engine, principally for the reason that blow holes may occur in the counterweight that would not otherwise be detected. When space is limited, as in the crankcase of a two-cycle engine, it is a very good scheme to core out the inner part of the weight and fill it with lead. The best method is to make the casting heavy and to drill the weights, to bring them to a balance.

In multiple-cylinder engines, other rules apply. Two-cylinder four-cycle engines are usually made with the cranks on the same side of the crankshaft, in order to make the impulses come evenly. Such engines must be balanced by counterweights in the same way as a single-cylinder engine. Three-cylinder engines of both the two-cycle and the four-cycle types are usually made with the cranks 120 degrees apart. Four-cylinder, two-cycle engines have the after cranks at 180 degrees from each other, and the forward cranks at 180 degrees from each other; but the two pairs are placed at 90 degrees. Hence, when the engine is running, there is an impulse for each one-quarter revolution. In the four-cylinder, four-cycle engine, the two center cranks are together, and the crank pins at the ends are at 180 degrees from those in the center. See Figs. 5 and 6.

With the exception of the two-cylinder, four-cycle engine with the cranks together, it is necessary that the pistons and connecting-rods should weigh very closely the same. The allowable variation should not be over ¼ ounce per pound. The crankshafts themselves should balance, and special attention should be given to the balancing of three-cylinder cranks, as this type is that most likely to be at fault.

It seems scarcely necessary to call attention to the balancing of the fly-wheel, but this is something often neglected, or done in a haphazard sort of way; and there is nothing that will make an engine vibrate more than a poorly balanced fly-wheel. As an example of what may be done in balancing, the writer has seen a four-cylinder, two-cycle engine running at 600 revolutions on which was standing on end a full-length pencil. This pencil was the ordinary Eagle diagraph, from which the eraser had been removed.

One of the greatest nuisances of the old-style marine engine is the hot exhaust pipe. Various methods have been employed to overcome this objectionable feature and so prevent the possibility of burns. These, the operator knows, are only too frequent. The first method adopted was to cover the pipe with asbestos, or other heat-insulating material. Then gradually it dawned upon several designers that it would be possible to water-jacket the exhaust. At first the idea was merely to decrease the noise of the exhaust, and the earlier plan was to allow a portion of the outlet water to enter the exhaust pipe near the engine. This helped in muffling the exhaust, but was of little service, so far as preventing burns was concerned. Then came the water-jacketed exhaust manifold, which in some cases is extended so that the entire exhaust pipe is jacketed to the outlet into the sea.

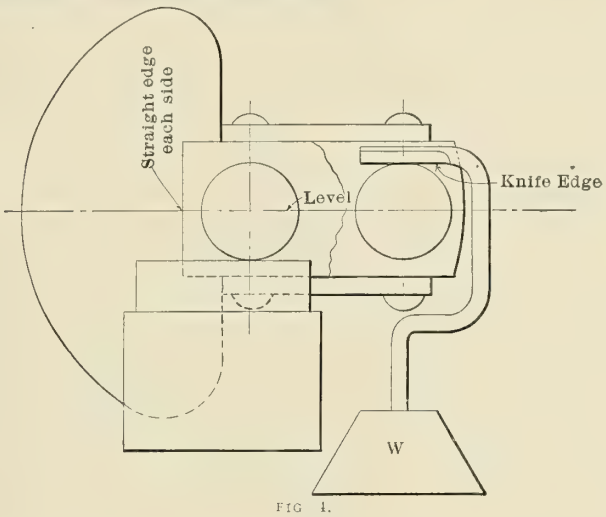


FIG 4.

The water-jacketed exhaust is easier attached to the two-cylinder than the four-cylinder engine. In certain styles of four-cycles, however, the exhaust is carried down through a pocket in the water-jacket of the engine and is brought out near the base. A hot pipe near the floor is not so apt to be troublesome as one within easy reach. If the exhaust is jacketed where it leaves the engine, and the pipe lagged with non-conducting material from where it leaves the manifold, it will be found quite satisfactory.

The usual practice is to lead the circulating water into the top of the cylinder and out through the jacket of the exhaust manifold. The opposite plan appears to the writer the more rational. An engine will run best when it is warm. On the other hand, it is desirable to keep the exhaust as cool as possible, both to cool the exhaust gases and thereby contract them, and to make it possible to touch the manifold without discomfort. It will be found that the warmer the engine is run, the greater will be its efficiency.

The location of the carbureter should be low, in order to have a sufficient head of gasoline to make the fuel flow freely.

The rule, on the average engine, is to so place the carbureter that the level of the gasoline in the float chamber is not over 2 inches above the center of the shaft.

Starting the small marine engine is a simple matter, but with the larger bores the task becomes an irksome one. The starting crank, to be effective, must be very long or geared. A later method is to use a long bar carrying a pawl to engage a ratchet on the crankshaft. The engine is turned over until an explosion takes place, or until just over the center, and the ignition current turned on. The latter is the safest method, as then a back kick will not carry the lever with the engine. In any case, some safeguard should be used to prevent the lever from being thrown backward and injuring the operator or damaging the boat. One builder of large engines makes the ratchet of cast iron, with weak teeth, so that in case of a back kick the teeth are broken. Three or four spare ratchets are carried on board. A much simpler way is to make the pivot pin of the pawl just strong enough to pull the engine over. A sudden jerk will shear the pin and prevent damage. It is much cheaper and simpler to replace, and takes up less room than the ratchet. In fact, an ordinary wire nail will answer for the pin. In Britain, the starting lever is mounted on an A-frame, at an easy position for the operator. This method has not found favor in America.

For large engines, the more rational way of starting is by means of compressed air. In fact, when the horsepower runs into the hundreds it is by far the most practical. The simplest way to start on compressed air is to use some form of throttle valve, and to admit the air by hand at the proper time in the stroke. A later improvement is to employ an air valve operated from the cam shaft. The engine then operates as an air engine until the gasoline cycle is taken up. In order to get an impulse every revolution, the cams are so arranged that, in starting, the mixture inlet valve is closed, and the exhaust valve opens at each revolution. This is usually arranged so that, in a multiple-cylinder engine, one cylinder is running on air while the others are taking up the gasoline cycle.

The latest and the best method, from the practical standpoint, of operating large engines is to so construct the engine that it can be run either ahead or astern by shifting the cams. The air-starting mechanism is arranged to work in both directions, and the engine can be handled very much like a steam engine. The mechanism used to operate an engine ahead or astern is very complicated for a four-cycle engine, and is hardly worth while on engines under 100 horsepower. For the smaller engine, reversing propellers and reversing gears are used, with the latter somewhat in better favor. A few two-cycle engines are arranged to reverse by means of the ignition. This is quite satisfactory for small craft, provided the reversing mechanism is made automatic and controlled by a governor.

The growing favor of the two-cycle motor, especially for the smaller craft, calls for a few remarks on the design of this class of motor. For many years we have been accustomed to the old two-port type, in which the suction of the charge into the crank case is through a check valve. For speeds below 600 revolutions per minute this type has been very satisfactory; but when running at higher speeds, the three-port type gives the best service. With the advent of the racing boat, growing more and more into favor each year, the demand was for comparatively light motors of the high-speed type. There at once came a protest from the builders of low-speed motors that good service, especially in the two-cycle, could not be obtained at high speed, and that the motor would be short-lived. The assertions they made were, however, based upon their trials with low-speed motors at high speed, and the fact was overlooked that we have had high-speed steam engines with us for a number of years, giving good service. Gasoline engines will stand up just as well when operated at high speeds as at low speeds, provided that they are correctly proportioned for the speed at which they are to be run.

The main points to consider, when designing an engine to run at high speed, are to make the reciprocating parts as light as possible without sacrificing strength, and to make the bearings of ample length. The crank pin, being the hardest to lubricate, should have particular attention. The designer is especially asked to note that it is the length, rather than the projected area, of the bearing surface that is important in the

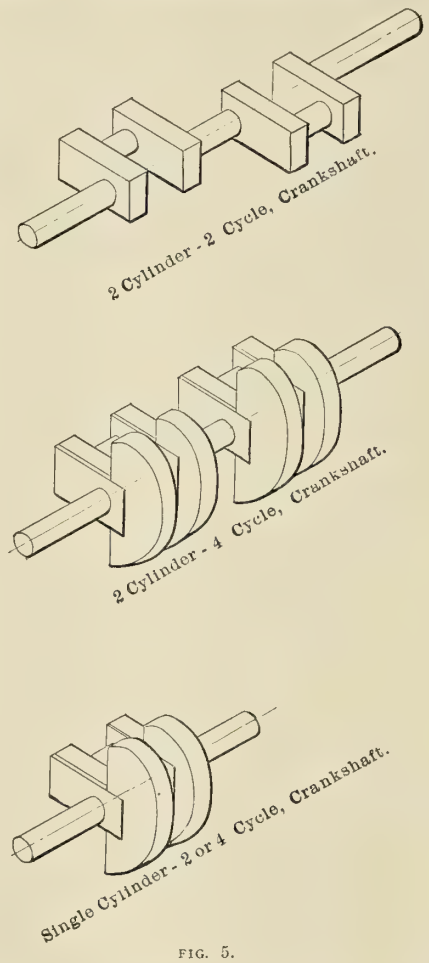


FIG. 5.

design of high-speed engines. In his *Manual of the Steam Engine,* the late Professor Thurston gives the following formula for the length of an engine bearing:

Let P = the total mean pressure = area × mean effective pressure,

R = revolutions per minute,

L = length of the bearing in inches.

$$\text{Then } L = \frac{P R}{600{,}000}$$

It has been found, however, in high-speed gasoline engines, especially where the lubrication was the least uncertain, that the formula given is very likely to cause trouble with hot crank pins. In recent designs a constant of 500,000 has been used for the denominator of the above fraction, and even as low as 400,000, where circumstances would allow. This extra length is especially desirable in high-speed, two-cycle motors, wherein the oil is swept from the base almost as fast as it enters. For example, the constant 500,000 gives for a 4-inch bore and a mean effective pressure of 80 pounds per square inch a length of 2 inches at 1,000 revolutions per minute. For a 4½-inch bore, the length of the pin should be 2½ inches.

In the length of the crankshaft bearing, the designer is

usually restricted by the length over all of the motor. The inner bearings between the cylinders should be approximately twice the diameter of the shaft, and the outer bearing at least two and one-half times the diameter of the shaft, especially in a high-speed, two-cycle motor.

The length of the piston pin can usually be made the same as that of the crank pin, and, in order to insure the minimum amount of wear, the pin should be fully this length, or at least as long as the piston will allow, when it has to be made shorter. It is necessary always to get a good bearing in the piston. This does not mean approval of fastening the pin in the rod and allowing it to turn in the piston, as it is preferable to place the bearing in the connecting-rod.

Short pistons in high-speed engines wear loose quickly and, when this happens, they are inclined to slap. Many designers, especially of automobile motors, turn their pistons small at the center and thus reduce the wearing surface, under the mistaken impression that in this way they reduce the wear. It is usually considered a well-established principle in design that, for the same conditions, the larger the bearing surface the longer the wear. The piston should be not less than one-third longer than the diameter of the cylinder.

High-speed, two-cycle motors, carefully designed, give astonishingly good results in power and speed. The power of a well-designed two-cycle, three-port engine will be 1.32 delivered horsepower for each 100 cubic inches of piston displacement per 100 revolutions per minute. The best record of a four-cycle motor is 0.98 horsepower, and a great many give only 0.88. It is reported that cases of both two-cycle and four-cycle motors would do better than this, but the information was not such that the accuracy of the figures could be guaranteed. The figure 0.98 for the four-cycle motor has been obtained both from large motors operating at moderate speed and from small motors operating at high speed. For instance, 15.4 horsepower has been obtained from an 8 by 10-inch cylinder at 330 revolutions per minute, and 12 horsepower from a 5½ by 6-inch cylinder at 900 revolutions. With a mechanical efficiency of 85 percent, this would mean a mean effective pressure of 89 pounds per square inch. With the two-cycle, the mean effective pressure by comparison is but 60 pounds, due principally to the fact that a considerable portion of the stroke is employed in emptying and recharging the cylinder.

For some time it has been the practice of builders of marine gasoline motors to rate them at about 75 percent of their actual power. This is confusing, and is apt to lead to many misunderstandings. Since a marine engine of this type is nearly always run at its full power, there is no good reason why it should not be so rated. Therefore, a 5½ by 6-inch, four-cylinder, four-cycle should be rated at 48 horsepower, if designed for a high-speed motor, and proportioned according to the speed at which the motor is to be run. This size, properly designed, can be depended on for 48 horsepower at 900 revolutions.

The speed at which a marine motor should be run depends somewhat upon the boat and the class of water. Speed boats pure and simple need high-speed motors, and the higher the speed of the boat the higher should be the speed of the motor. This is for two reasons. First, in order to get the requisite power, without overloading the boat, a slow-speed motor may not be used. In speed boats, they being usually of shallow draft, the propeller should be of small diameter and comparatively low pitch. In order to get a high speed with a small pitch, the speed of the propeller must, of course, be high.

In slower boats it is comparatively easy to operate with a high-speed motor using a propeller of small diameter and pitch, provided that the boat does not have to buck into extremely heavy weather. Small wheels, however, have not the area to thrust against heavy resistance.

There is a great deal in the choice of a propeller wheel suitable for not only the engine, but the boat as well, and it is a good thing for the designer to have reliable data at hand on this matter. For example, identical engines will turn a certain wheel not over 800 revolutions per minute in a family launch, and in a racer will drive it up to 1,100 revolutions. The propeller for a speed boat should be most carefully balanced, otherwise it will not only cause the engine to vibrate, but it will reduce its speed. Ordinarily, the three-bladed

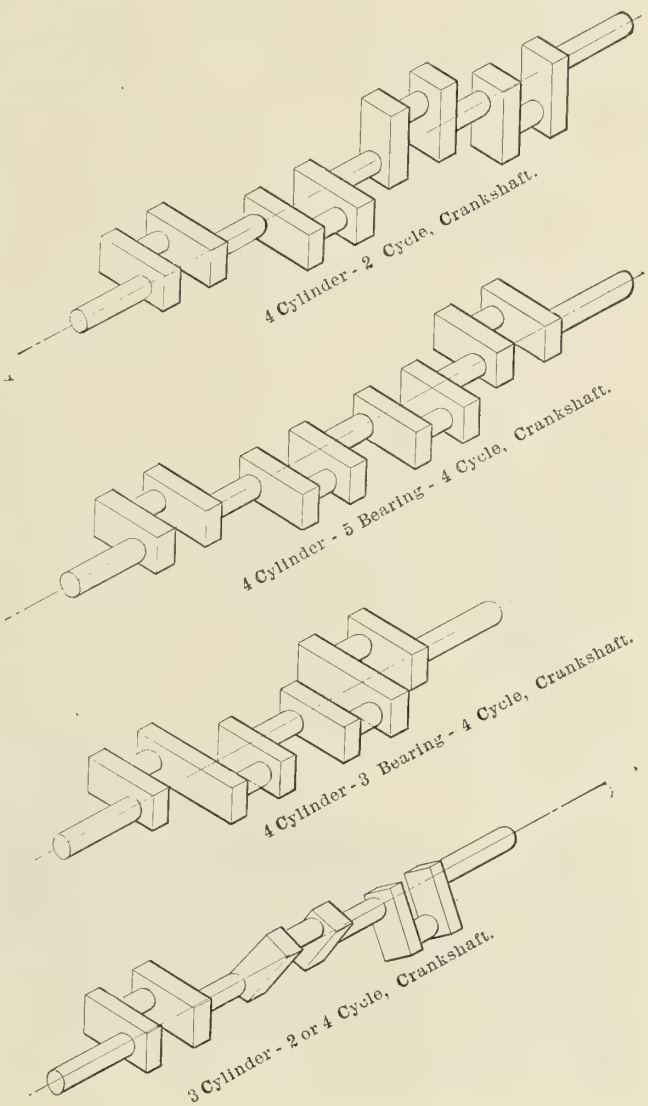

4 Cylinder - 2 Cycle, Crankshaft.
4 Cylinder - 5 Bearing - 4 Cycle, Crankshaft.
4 Cylinder - 3 Bearing - 4 Cycle, Crankshaft.
3 Cylinder - 2 or 4 Cycle, Crankshaft.

FIG. 6.

propeller has been found to be better balanced than the two-bladed. Two propellers have been taken, one a two-bladed and the other a three-bladed, both alike in diameter and pitch, and tested with the same boat and the same motor. With the three-bladed, the engine ran 720 revolutions per minute, and with but little vibration; while with the two-bladed, the engine slowed down to 650 revolutions, and the vibration was so great that the power had to be shut off.

Not being a propeller expert, the above experience with the two and the three-bladed screws is given for what it is worth. The propellers were furnished by a propeller specialist, and his word was taken that they were carefully balanced. Yet other propeller makers state that it is possible to make a two-bladed screw that will run as well at high speeds as one with three blades.

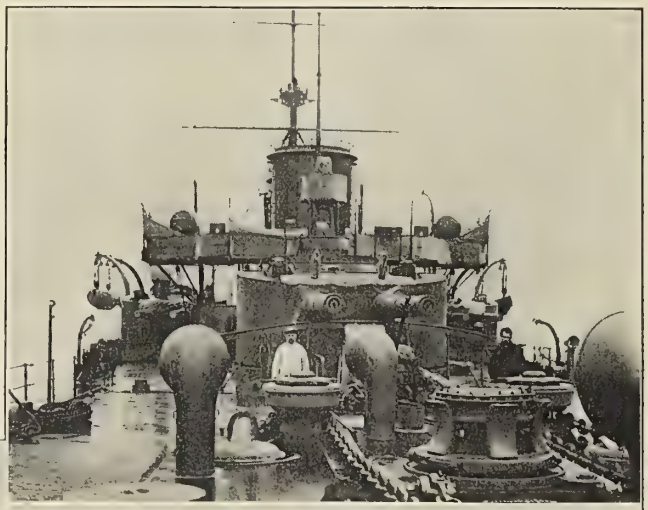
The Russian Cruiser-Battleship Rurik.

BY BENJAMIN TAYLOR.

We present a final photograph of the Russian cruiser-battleship *Rurik,* taken during her last series of steam trials on the Clyde, on the completion of which she was handed over to the Russian government—a completed bargain. We also give a deck view, taken from the bow of the ship, showing barbette, etc.

The speed trials were made in several runs over the measured mile at Skelmorlie, Clyde. The condition was that the engines should keep up an average number of revolutions equaling or exceeding the number corresponding to 21 knots, as ascertained on the measured mile trials. This was easily accomplished. The *Rurik* also carried through two 30-hour trials, one at about 19 knots and the other at about 12 knots. These trials were to determine the radius of action.

The mean of 10 hours at full power during 24 hours' trial was: Steam pressure, 280 pounds per square inch; vacuum, 26.2 inches; revolutions, 141.6 per minute; indicated horsepower, 20,675. All her twenty-eight (Belleville) boilers were in use, and the air pressure in the stokehold averaged 0.36 inch. The guaranteed horsepower was 19,700, and the revolutions required were 135 for 21 knots, so the results exceed the contract; the speed maintained having been approximately 22 knots. The heating surface is over 55,000 square feet.



THE RUSSIAN CRUISER-BATTLESHIP RURIK.

VIEW FROM FORECASTLE LOOKING AFT.

Tests were also made of coal and water consumption, and as to an important condition in the contract with reference to stability. The temperatures were tested in various parts of the ship, etc., and maneuvering trials were carried out satisfactorily.

The *Rurik* was designed for a displacement of 15,200 tons on a draft of 26 feet, and is well within these limits. Her length between perpendiculars and on waterline is 490 feet, and the molded breadth is 75 feet. On normal displacement she carries 1,200 tons of coal, which may be increased to 2,000 tons by filling the bunkers. With the latter quantity, the estimated steaming radius at 10 knots is 8,000 nautical miles.

The machinery has been designed less for economy of weight than reliability. Her boilers are fitted for consuming coal or oil fuel. The propelling machinery consists of two sets of four-cylinder triple-expansion engines. The power, under easy steaming, is 20,000 indicated horsepower, with the boilers working at a steam pressure of 285 pounds to the square inch, reduced to 250 pounds before entering the engines. A complete system of auxiliary machinery has been fitted, and the pumping and drainage arrangements are the outcome of Russian Admiralty plans.

In respect to her armament, the *Rurik* is remarkable because, though called an armored cruiser, she has all the fighting power of a battleship. She is also specially equipped for repelling torpedo-boat attacks. All her guns have, possibly, a greater elevation and depression than in other ships. She has four 10-inch breechloading guns, 50 calibers in length, twin mounted in barbettes forward and aft on the center line, and each of them can be worked through 35 degrees of elevation and 5 degrees of depression. The forward guns can be trained 45 degrees abaft the beam, and the after guns 45 degrees before the beam. She has also eight 8-inch breechloading guns, 50 calibers in length, twin mounted in barbettes on the quarters of the ship. All these guns are electrically worked.

For repelling torpedo attack there are twenty 4.7-inch quick-firing guns of 50 calibers in length; sixteen of these are in an armored battery in the center of the ship, separated from each other by transverses of specially hardened armor. This battery enables the secondary armament to be placed higher above the waterline than usual. It incidentally adds largely to the armored protection, as it is above the normal armored

side of similar ships. There are four 4.7-inch guns aft, also within armor on the sides of the ship. To counteract the effects of raking fire, three armored bulkheads have been fitted. There are twelve smaller quick-firing guns. Two 18-inch torpedo tubes, completely submerged, are placed forward.

On the gun trials 100 rounds of ammunition were fired from one of each type of gun. Thirty rounds were fired from two of the 10-inch guns and two of the 8-inch guns, and fifteen from the other guns, at various angles of elevation and depression and on various bearings. The gun mountings were worked to give a rapidity of two rounds per minute from the 10-inch guns, three rounds from the 8-inch guns, and eight rounds from the 4.7-inch guns. The 47-millimeter (3-pounder) guns were fired at the rate of between twenty and thirty rounds, and the Maxim guns at about 500 rounds per minute.

The *Rurik* has a complete armor belt from end to end and to a considerable distance below the waterline. It is 6 inches in thickness, tapering to 4 inches (bow) and 3 inches (stern) at the extreme ends of the hull. This belt is 12 feet deep, and the thick portion amidships extends over a length of 270 feet. For protecting the 4.7-inch guns, the upper strake, for 200 feet of the length of the ship, is 3 inches specially hardened armor. The barbettes within this central battery are of much heavier armor, having 7¼-inch walls. The conning towers are of 8-inch armor. Two range-finding towers extend a considerable height above the upper deck, and are constructed of 5-inch armor. Protective decks of a combined thickness of 4 inches are arranged, to insure that high explosive shells shall burst outside the vitals of the ship. The base of each of the three funnels is protected by armored casings. The whole of the machinery and magazines are, under the waterline, surrounded by armored walls extending from the main deck, through the protective deck, to the bottom of the ship.

The *Rurik* was built, engined and armored by the Vickers Sons & Maxim Company, Barrow-in-Furness, having been launched Nov. 17, 1906.

The Russian papers have been severely criticising the construction of the ship, asserting that during recent firing tests on board, conducted by a special commission, four out of the eight 8-inch guns constituting part of the main battery were rendered unserviceable by the sinking of the turret foundations. It is declared that the cruiser will be valueless as a fighting unit until the turrets are rebuilt. The English constructors of the vessel attribute this to the faulty designs furnished by the Russian Technical Committee, while the Admiralty blames the builders.

A comparative table is given of the latest ships of this general character belonging to the seven leading naval powers. The only broadside greater than the *Rurik's* is that of the (battleship) *Kurama.* The propulsive results on the *Rurik,* as shown by the Admiralty coefficient, are remarkably fine; while the French and German designs show poor results.

	Russian, *Rurik.*	British, *Minotaur.*	American, *Montana.*	French *Rousseau.*	Italian, *Napoli.*	German, *Blucher.*	Japanese, †*Kurama.*
Displacement..	15,200	14,600	14,500	13,644	12,426	14,760	14,636
Horsepower....	20,675	27,856	27,489	*37,000	*23,000	*35,000	*22,000
Speed, knots...	22	23.01	22.26	*23	*22	*23	*21.25
Admiralty coef.	316	261	239	188	248	209	261
Length, feet....	490	520	502	515	470	499	450
Battery.....	4—10″ 8—8″ 20—4.7″	4—9.2″ 10—7.5″	4—10″ 16—6″ 22—3″	14—7.6″	2—12″ 12—8″ 12—3″	12—8.2″ 8—5.9″ 20—3.4″	4—12″ 8—8″ 14—4.7″
Broadside, lbs.	3,450	2,520	2,954	1,665	3,032	2,532	4,715

* Estimated. † Really a battleship.

THE SEA-GOING TUG DARENT, BUILT BY FERGUSON BROTHERS, PORT GLASGOW.

A New Thames Tug-Boat.

The powerful screw tug *Darent,* built by Ferguson Brothers, Port Glasgow, to the order of the Thames Conservators, carried out her speed trials on the measured mile, the mean speed of six consecutive runs being 11¾ knots, considerably in excess of the contract, and highly satisfactory.

The vessel is intended for general harbor service and to act as a tender to the conservancy dredging fleet. The officers' and crew's quarters are arranged aft of the machinery space, and are spacious and well equipped. The saloon forward is paneled in polished oak, the pantry and lavatories being tiled on floor and walls. A roomy chart house is fitted under the bridge deck.

Davis steam steering gear is fitted on bridge, also Chadburn's telegraph, with speaking tube and telephones to various parts of the vessel. Electric light by Clarke, Chapman & Company is fitted throughout, with powerful cluster lights for salvage work. The forward windlass is of Harfield's make.

The propelling engine is of the triple-expansion type, fitted with Brown's patent reversing gear, Axiom lubrication and United States metallic packing. Steam is supplied by a large multitubular boiler having a working pressure of 180 pounds per square inch.

The following auxiliaries are fitted in engine room: one

powerful fire and salvage pump, with swivelling monitor on deck, Caird & Raynor's evaporator, Railton & Campbell's feed filter, automatic float tank, feed pump, electric generator with direct-acting engine, and a fresh-water duplex pump. Fresh-water compartments are arranged in the vessel for supplying the dredgers, barges, etc., on the Thames.

Throttle Watch in a Typhoon.

BY S. F. SMITH.

A few days after target practice, on a hazy, hot, tropical morning, in Manila Bay, the word was passed by the boatswain's mate, "All hands up anchor!" The windlass rattled, and the "pick" was soon at the hawse-pipe.

The navigating officer on the bridge gave orders to port the helm, and moved the pointers of the engine-room telegraph from "stop" to "slow ahead" on starboard; to "slow astern" on the port. The 13,000-ton battleship trembled slightly from the motion of her engines, and began to swing slowly and majestically around to port, pointing her bow towards Corregidor Island, bound for Hong Kong, where we were to go into the Kowloon dry-dock to be cleaned, painted and have our sea valves overhauled.

We passed the historic Corregidor, and steamed up along the western coast of Luzon at a 10-knot speed. The weather continued hot, and the sea was calm, just a light air from the south, directly astern, causing poor draft, and hot, hard work in the fire rooms, in order to keep up the prescribed 150 pounds of steam at the boilers. The thermometer in the fire rooms registered 168 degrees F., in the engine room 138 degrees F.

The firemen, between the acts of firing, slicing and pushing back, dove under the ventilators and mopped the streaming perspiration from their eyes. "Whew! She's hot." But there is not much time to think about it—the above-mentioned cycles come around too often. No, the "land-lubber" fireman couldn't live down there.

About four bells, or after the evening meal, the wind changed to the eastward, and a slow, drizzling rain started. Someone said the "glass" was falling rapidly, indicating "a night of it." The men began to array themselves in their oilskins and "sou'-westers," and collected in small groups to the lee of the forward 13-inch turrets, smoking their pipes and talking in low tones about the prospects of the coming night, and listening to the prophecies of the older tars, who had had various kinds of "runs" through these seas before.

It was now seven bells, and raining hard, and the wind was blowing in gusts, causing the ship to list over to port a few degrees. It also caused the men around decks to seek shelter of some kind. We were now leaving the northern end of Luzon, and every now and then Cape Bojedor light—which flashes every minute—could be seen a few points off our starboard quarter. We began to feel the heave of the ship more distinctly now, as we plowed out into the Yellow Sea.

My hammock was swung on the berth deck. On my way to it from the main deck I had to pass through the gun deck. On entering this, and looking around, I found a far different scene from that on the main deck; men and boys were lounging around on their ditty boxes, playing cards, reading, talking and laughing, and it soon made one forget the dark night and the storm brewing outside.

No sooner had I rolled into my hammock than the whistle of the boatswain sounded, and after it, "Relieve the wheel and lookout!" I settled myself comfortably for my four hours' sleep; eight bells strike, the whistle again and after it, "Turn in your hammocks and keep silence about the decks!" was the last I heard before going to dreamland.

The next thing I knew someone was shaking me, saying, "Roll out, there; fifteen minutes, roll out!" meaning that I had only fifteen minutes to "roll out," dress and relieve my man in the engine room. I was just begining to think he was never going to stop shaking me, but on looking around I found everything else was being shaken up, from the ship "wallowing into it," so I "rolled" and hastily got into my "watch clothes"—not one of the shining uniforms the uninitiated "land pikers" see us in on our trips ashore—but overalls and a sleeveless shirt and a pair of No. 8 regulation navy shoes, which I sorted out from a half dozen other pairs, all having slid down to leeward against the bulkhead. I also picked up my cap from among them, and, after adorning myself in these things, staggered out through the alleyways to the engine-room hatch.

Steadying myself on the hot hand-rails, I made my way down through the starboard battle hatch, over the top gratings, passed the high-pressure cylinder—which greeted me with the familiar sound of heavy breathing as the steam entered and exhausted from it—down the ladder to the lower or handling platform.

The telegraph pointer stood at "half speed ahead," and the engines were running to correspond. I made the usual rounds before relieving my man at the throttle, and then went up to him, saying, "How is she, Peters?"

"Oh, everything is all right; we have been running half speed about two hours; I guess she is pounding into it pretty hard up there; steam 125, revolutions 55." And he went up the ladder, seeming glad his four hours of nerve strain was over, and little thinking he would be turned out to the tune of "All hands save ship!" inside of two hours more.

I took my station at the various levers by which the speed of the 5,500-horsepower engine is controlled, counted the revolutions of the high-pressure crank, noted the readings of the steam, receivers and vacuum gages and the indicator on the bulkhead, which divides the port from the starboard engine room, and found my engine leading slightly, which was proper. The warrant machinist went from one engine room to the other, not saying a word; but by the expression of his face I knew all was well.

The ship was pitching heavily, causing the engines to race slightly when settling her stern into the trough of the sea. At these moments I choked her with the butterfly valve, and outside of that the regular engine-room routine was carried on for the rest of the hour.

At one in the morning (two bells) the telegraph jangled, and the pointer came to rest at "Slow ahead." I choked her down to 30 revolutions per minute, and slowed down the main air and circulating pumps, and notified the water tender to check firing. The ship was now laboring heavily, and great care was necessary in moving around on the slippery floor-plates, or the result would be serious.

Everything was again going smoothly after slowing down. The ship did not receive the hard shocks as before, but rolled and pitched more heavily; so it was only to hold on, keep ears and eyes open, and watch the rhythmic rise and fall of the cranks, with their occasional racing. As this became monotonous, I whistled to the fire room, asking the water tender if all were well out there. The answer came back, "All O. K. out here."

I then asked Wilson, in the port engine room, how things were over there. "Oh, it's just lovely over here; I'm hanging on with my teeth and hands. I guess this must be a typhoon, all right."

"I guess it is the"—whew! whiss! hiss! hiss-s-s! Water comes down the hatch and ventilators, striking the hot pipes and cylinders, partly turning to steam. I try to duck it, with poor success; the ship takes a heavy dive, everybody holds on, and looks at each other. More water comes down the ventilators; the telegraphs jangle briskly, and the pointer stops at "Very slow ahead." I choke her, and notice the electric lights are getting dim, with occasional "winks."

"What do you think is happening?" from the young messenger boy.

"I don't know, sonny," I answered, and told him to find a grease lamp and light it, but he seems so scared that he does not dare leave me. The lights now slowly die out entirely; something must be wrong with the dynamos. I told the oilers to light the bulkhead oil lamps with their hand lamps.

The water tender comes from the fire rooms. "Nos. 5 and 6 fire rooms are flooded! The coal has washed off the floor plates into the bilges and choked the strainers, and the pumps won't work on them!"

"Do the best you can to clear them, and get the pumps going on them." The warrant machinist goes with him to help keep the men in order.

A 'phone message from the bridge asks us how things are down here. We answer, "Our lights are out, the forward fire rooms are flooded, and the strainers choked with coal."

"Do the best you can," from the bridge. The chief engineer is down here now. He goes to the fire room, and comes back with the warrant machinist.

The engine cranks are now dipping into the water as it comes rushing in from the fire rooms, for the small bilge pump cannot handle it all; so we swing the main circulating pumps from the sea suction to the main drains.

I 'phone to the dynamo room, telling them our lights are out. "We are flooded up here; the water came down our ventilator and wet all our machines, so we had to shut down," comes the answer.

The word has long ago been passed on deck to "All hands save ship!" and bucket gangs are formed to clear the dynamo room and other compartments of water; some are at the hand pumps. The main drain strainers have choked with coal from the fire rooms, and we swing the main circulating pumps back on the sea suction, in order to cool our condenser, for the vacuum is falling.

We send men "diving" down to clear the strainers and try the pumps again, and so on for an hour or more, until we finally clear the bilges of water; then thoroughly clean out the coal and replace the floor plates. The forward fires are again worked, and general order is restored throughout the fire rooms and the engineering department in general.

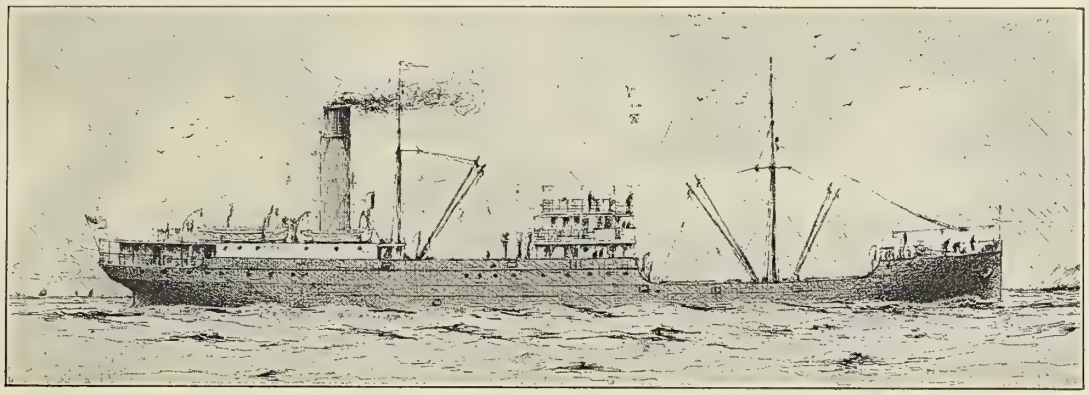
It is now 3 o'clock, and we are still under "grease lights." I telephone to the dynamo room: "How soon can we have lights?"

"We don't know, as there is a great deal of water washing around the place yet."

My! but it must be nice up there; hot! whew! The dynamo room is situated directly over the boilers, and has only one 10-inch ventilator, through which they draw air with an electric suction blower, and that is now stopped on account of no "juice" from the dynamos, and only grease lamps for light, and the ship laboring in the midst of a bad typhoon.

Try to imagine yourself in a corner of this room, hanging on with one hand and bailing water into buckets with the other; catching the water on the jump as it comes down towards you with the roll of the ship, dodging oil cans, waste cans, tools and no end of things that break loose in the excessive rolling. Oh, no! you don't have time to think of home and dry land, or if the ship is going down; things come too swiftly, and there are too many orders to execute.

The ship is now "hove to," and we ride it out. Two days later "Jack" is in the land of the Celestials, enjoying himself, and—never a typhoon.



THE STEAMER PARINGA, BUILT AT KINGHORN FOR AUSTRALIAN COASTING SERVICE.

An Australian Coasting Steamer.

The steamship *Paringa,* built by Messrs. Scott of Kinghorn, Ltd., was launched in April last for the Adelaide Steamship Company, of South Australia. The dimensions of the vessel are: Length between perpendiculars, 230 feet; breadth, molded, 36 feet; depth, molded, 16 feet 4 inches, with a gross tonnage of about 1,300. She has been built to the highest class with the British Corporation and to British Board of Trade requirements for a foreign-going passenger steamer.

This vessel has a long poop and forecastle, three cargo hatchways and five steam winches. Steam steering gear is placed aft in a steel house, and works direct on to the screw steering gear, which is of extra strong make, by Donkin & Company, Newcastle-on-Tyne. A large warping winch is also placed in this house.

The arrangement for working cargo is of the special design fitted on most of the Adelaide Company's vessels. A large casting is fixed to the deck, half-way between the center line and side of the ship. The goose neck for the derrick sits into a cast-steel nut, which can be moved by means of a square-thread screw. The head of each derrick is supported by chains to outriggers from the mast. The foot of the derrick can be moved in or out to suit loading or discharging cargo; and lifts are arranged for 10, 8 and 5 tons. There is also provision for fitting a 20-ton derrick.

Accommodation for twenty-eight first-class passengers is provided in two-berth rooms at the fore end of the poop on main deck. The rooms are tastefully fitted, and special attention has been made for keeping the rooms cool by means of large Boyle ventilators; each room is also provided with a goose-neck ventilator. These rooms are finished in white enamel and polished teak pilasters. The panels are polished jalousie, with expanded metal panels at top and bottom and an opening glass panel below the beams.

The dining saloon is in the deck house at fore end of poop. It is in polished hardwood, tastefully designed, with ceiling done in white and gold. Two electric fans keep this room cool. A smoke room is situated above, and is in polished hardwood, with ceiling of lincrusta, white and gold. An

electrically-operated fan is also provided for this room.

Accommodation is provided aft for sixteen second class passengers. The rooms are similar to the first class, jalousied and finished in white enamel. The captain and officers are berthed amidships on the lower bridge, while the engineers and petty officers are aft in sidehouses. The crew are forward in forecastle, and have their own bath room.

Special attention has been paid to all the accommodation, and all the rooms are large, airy and well ventilated, with everything that can add to the comfort of passengers and crew. Electric light has been fitted throughout the ship.

The engine, which is placed aft, is triple expansion, with a working pressure of 180 pounds per square inch. The cylinders are 21, 35 and two of 37 inches by 36-inch stroke. Two Babcock & Wilcox watertube boilers have been fitted. The grate area of the furnaces is 156 square feet, and the heating surface 5,000 square feet. A complete auxiliary outfit is provided, consisting of electric light plant, large ballast pump, sanitary pump and general steam pump.

The vessel was launched with steam up, and immediately proceeded to Burntisland under her own power.

Benjamin Taylor.

Trials of the Marine Contractor.

There was a busy atmosphere about the low-ceilinged office, with its corps of young men, each lending his share to a well-managed, energetic shipyard, the clicking of the typewriter following the earnest dictation (out of a cloud of cigar smoke) from the ever-busy manager, whose eyes gleamed at the prospective business this bid would bring, and the excitement that comes from honest competition to secure contracts from sealed bids. Most of us have felt the keenest nervous strain at one time or another in our lives, whether it has been with rod and reel, with the gun in a quiet valley, thick with undergrowth, out of which the gamest of birds start for liberty with a thrilling roar, or with other sports or anxious moments; but there is hardly a more anxious moment, when one's nerves are more highly strung, than that when the authority says: "Time is up; any more bids? Bids are closed." Then the opening.

A strange picture is presented at these openings. Seldom does the man whose money is at stake appear; more often one not so deeply interested represents the firm, and even among these men, of all ages and types, there is a grand opportunity for the study of face contortions. It is doubtful whether the actual contractor, having been through these exciting moments when he was assistant to another, could stand the strain of these ordeals, which are part of the daily business routine. They are usually well and gladly rid of the mission, but stand close by the 'phone at their offices awaiting the result, which their lieutenant hastens to transmit when all bids have been read and the successful bidder announced.

It was for a tremendous repair job to a large vessel, that all were anxious to get at a good price; but, work being scarce at the time, the chances for cut-throat prices were propitious, and generally understood to be likely. We visited the vessel, worked nights on the estimate, which was asked for to be in and opened three days hence. This, by the way, is customary—to have the contractors crowd a good week's work into this time or less, and necessitates a nerve-racking task, trying first one way and another to arrive at a fairly good price likely to secure the job, but still have a fair margin of profit, and name the fewest possible days that will complete the work, always having in mind a large demurrage payable for each and every day one fails to finish the work.

The bids opened, we were declared successful and the contract was awarded. A groan of relief arose from everybody, but a sigh of anxiety from the "boss." Immediately orders, right and left; things must be set in motion; no time to be wasted; minutes mean dollars from now on; a good figure for the job and a short period to finish; $300 per day demurrage for each and every day over contract time. Three hundred dollars bonus for each and every day contract completed previous to contract time. After the best part of contract time, all of which was spent in good, hard labor, the books show a good profit in sight—work well in hand—looks to be easily completed within the time and a few days' bonus in sight.

A snag is struck; needless to go into details; must be overcome, and means a vast amount of additional work. The foreman erred; the superintendent beside himself to get out with least possible loss of time and labor; estimated profit gone and demurrage seems inevitable. Owners of the vessel become anxious; have arranged sailing, and delay means more to them than amount of penalty to be suffered by contractor; day for completion rolls by and finish not in sight. The great blunder carries blame for every man connected with the plant, and when the ship finally steams away there's no thanks, but "hell to pay" for all hands.

Such is the life that may be understood only when directly connected with the concern handling such a character of work, and yet should you chance to be a visitor on one of the worst or most exciting days of these struggles, there is not the slightest surface disturbance evident; such strong, manly hearts and constitutions are meted out almost providentially to those to whom such cares and worries are no more than is expected. Smooth sailing comes more often than such trouble, but experience teaches that in the times of healthy prosperity perversity must play its part.

The Fire Float Beta.

In our September issue appeared photographs and a description of the pumps fitted to this vessel, which was specially designed by F. J. Trewent & Proctor, London, E. C., for this purpose. The hull was built of steel by Forrestt & Company, Wivenhoe, and is 100 feet 6 inches in length by 16 feet 6 inches molded beam, with a draft of 3 feet 6 inches—a specially light draft being necessary to enable the vessel to get as near to the shore as possible when attending fires. There are six watertight transverse bulkheads, which, in conjunction with fore-and-aft bulkheads on each side of the machinery and boiler space, render the craft practically unsinkable in the event of damage.

The twin-screw triple-expansion propelling engines and two watertube boilers, each fitted with an automatic feed regulating device, were manufactured by Mumford & Company, Colchester. The boiler room outfit includes a pair of powerful automatic feed pumps and general service pump, an evaporator for supplying fresh water to the boilers and a powerful forced draft fan. Each of the two boilers is capable of running the propelling machinery at full power. Steam is therefore always kept on one boiler, so that the vessel can be dispatched at a moment's notice. While on her way to any fire, the other boiler can be lighted up and be under full steam in a very short time, so that on reaching her destination both boilers would have a full pressure of steam for driving the pumps, both boilers being required to work the four pumps at their fullest capacity. The boiler room is arranged on the closed stokehold principle, and fitted with a powerful fan for forcing in air under pressure.

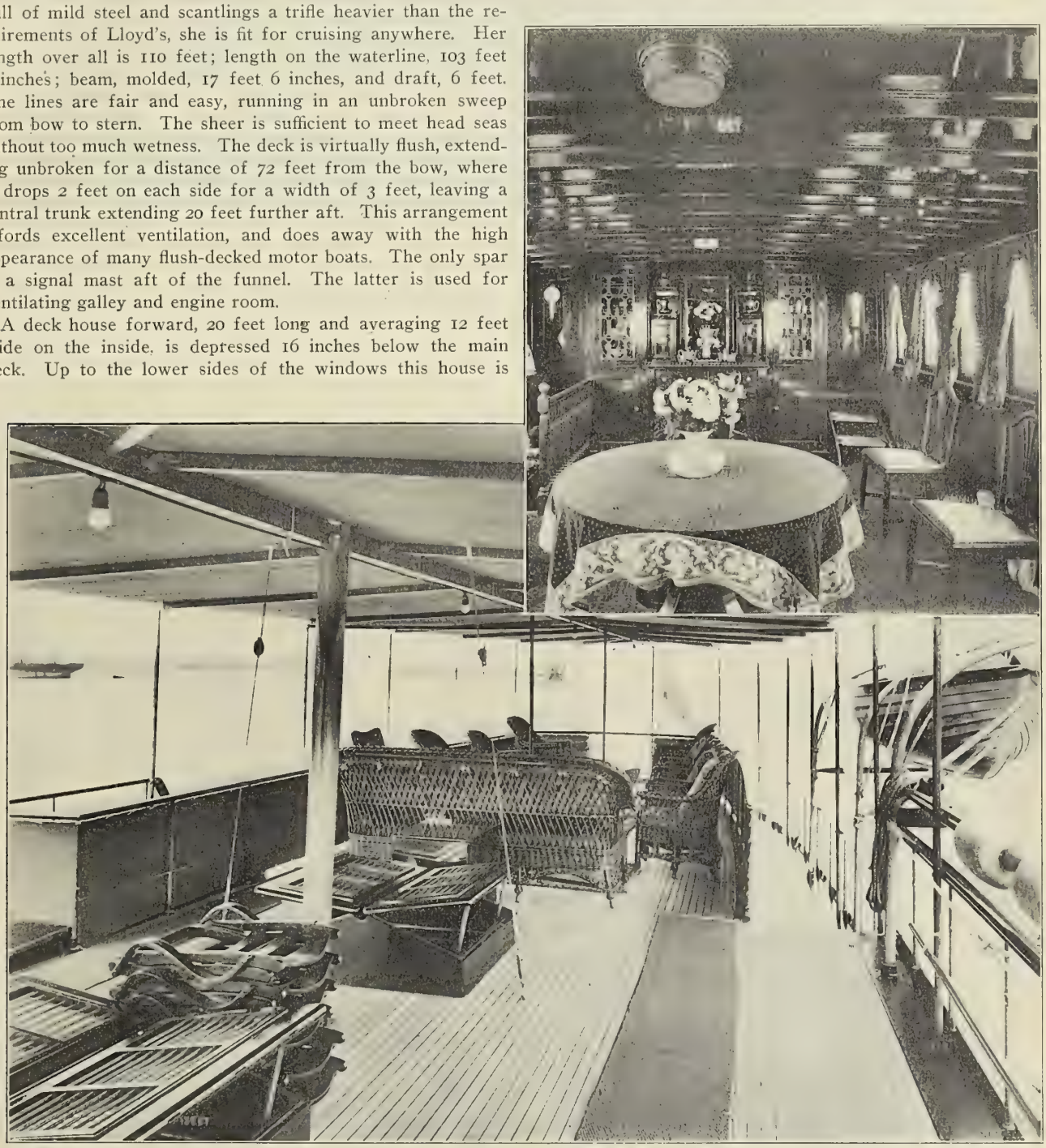
It is said that this vessel has the most powerful fire fighting installation of any afloat on such a light draft; and in designing her for this purpose it was found advisable to arrange the twin screws on the tunnel system, so as to obtain large enough propellers for the required speed. Suitable quarters are arranged for the crew aft, and she is lighted throughout with electric light.

A LARGE TWIN-SCREW MOTOR BOAT.

The *Cristina,* designed by Henry J. Gielow, New York, and built by George Lawley & Son, South Boston, for F. C. Fletcher, of Boston, is the largest of her type, and, having a hull of mild steel and scantlings a trifle heavier than the requirements of Lloyd's, she is fit for cruising anywhere. Her length over all is 110 feet; length on the waterline, 103 feet 5 inches; beam, molded, 17 feet 6 inches, and draft, 6 feet. The lines are fair and easy, running in an unbroken sweep from bow to stern. The sheer is sufficient to meet head seas without too much wetness. The deck is virtually flush, extending unbroken for a distance of 72 feet from the bow, where it drops 2 feet on each side for a width of 3 feet, leaving a central trunk extending 20 feet further aft. This arrangement affords excellent ventilation, and does away with the high appearance of many flush-decked motor boats. The only spar is a signal mast aft of the funnel. The latter is used for ventilating galley and engine room.

A deck house forward, 20 feet long and averaging 12 feet wide on the inside, is depressed 16 inches below the main deck. Up to the lower sides of the windows this house is constructed of steel plating, and is finished with teak panel work both inside and outside. This deck house is fitted up as a dining room, using ordinary chairs, and a round extension table. Aft on the starboard side a stairway leading down to the staterooms and cabin is visible at the left of our illustration. On the port side steps lead to the main deck. Between the two stairways is a sideboard and buffet, finished in teak, with a small dumbwaiter leading to the galley immediately below.

The only other obstructions on the flush deck are two skylights aft of the funnel, and lighting, respectively, the saloon and the passage between staterooms, the companionway forward for the crew and a steering platform, 7 by 8 feet, located between the deck house and the funnel. This platform is inclosed with a brass railing and covered with an awning.



DINING SALOON AND DECK SPACE AFT ON STEEL TWIN SCREW MOTOR YACHT CRISTINA.

(Photographs, N. L. Stebbins, Boston.)

Below decks the forepeak is occupied by chains. Aft of the collision bulkhead is the forecastle, 14 feet in length, including the crew's toilet and locker rooms. Berths are here provided for four men. Staterooms for the captain and engineer, fitted with wardrobes, etc., come next. Then comes the engine room, 12 feet in length, immediately beneath the dining saloon. This room is inclosed in steel watertight bulkheads, with all woodwork fireproofed and covered with asbestos, in order to reduce to a minimum all danger from fire.

THE STEEL TWIN SCREW MOTOR YACHT CRISTINA, DESIGNED BY HENRY J. GIELOW.
(Photograph, N. L. Stebbins, Boston.)

Aft of the engine room is the galley, fitted with stove, dressers, sink, dish racks and closets, with an ice-box and refrigerator (with a capacity of more than ½ ton of ice) and a gasoline (petrol) tank (capacity 2,000 gallons) just abaft the galley space. This tank, which is inclosed in a steel watertight compartment, is constructed of galvanized steel, with rivets and seams soldered. The capacity is such as to give the boat a cruising radius of 1,100 nautical miles at full speed, or 2,200 miles at 10 knots. On the starboard side is a passage leading from the saloon to the staircase up to the dining saloon.

The owner's quarters and staterooms for guests are located in the after half of the vessel. Immediately abaft the galley is the saloon, running the entire width of the hull, and 11 feet in length. Aft of this on the port side are two staterooms with a toilet room between them, while on the starboard side are the steerage and passage to the main deck, a stateroom with double berth and a bath room. Still further aft is the owner's stateroom, extending the full width of the vessel, and containing a double berth, two wardrobes and a bureau. The after end of the hull is allotted to general stores.

Propulsion is by twin screws, each actuated by a six-cylinder 100-horsepower engine, built by the Standard Motor Construction Company, of Jersey City. These engines are located forward beneath the dining saloon, and the long length of shafting makes it possible to place the axis of the propellers in a nearly horizontal line. The rudder is balanced, and the dead wood is cut away aft to such an extent as to make for easy handling. The maximum speed is about 13 knots.

Three boats are carried, one of which is itself operated by a small gasoline engine. There is an electric light plant in the engine room, and the boat is heated throughout by steam. Running water is supplied throughout the entire vessel, a fresh-water tank with a capacity for 100 gallons being located in the general store room aft. The bath room and toilet room floors are tiled, as are also the sides and bulkheads of these compartments.

Our illustrations will show that the vessel is most tastefully fitted up, and that a prime requisite in the design was the subject of comfort. The expanse of deck space aft of the funnel is such as to give plenty of room for lounging and for such entertainments as may be desired by the owner. The boat was launched July 31 last, and has already seen some little service.

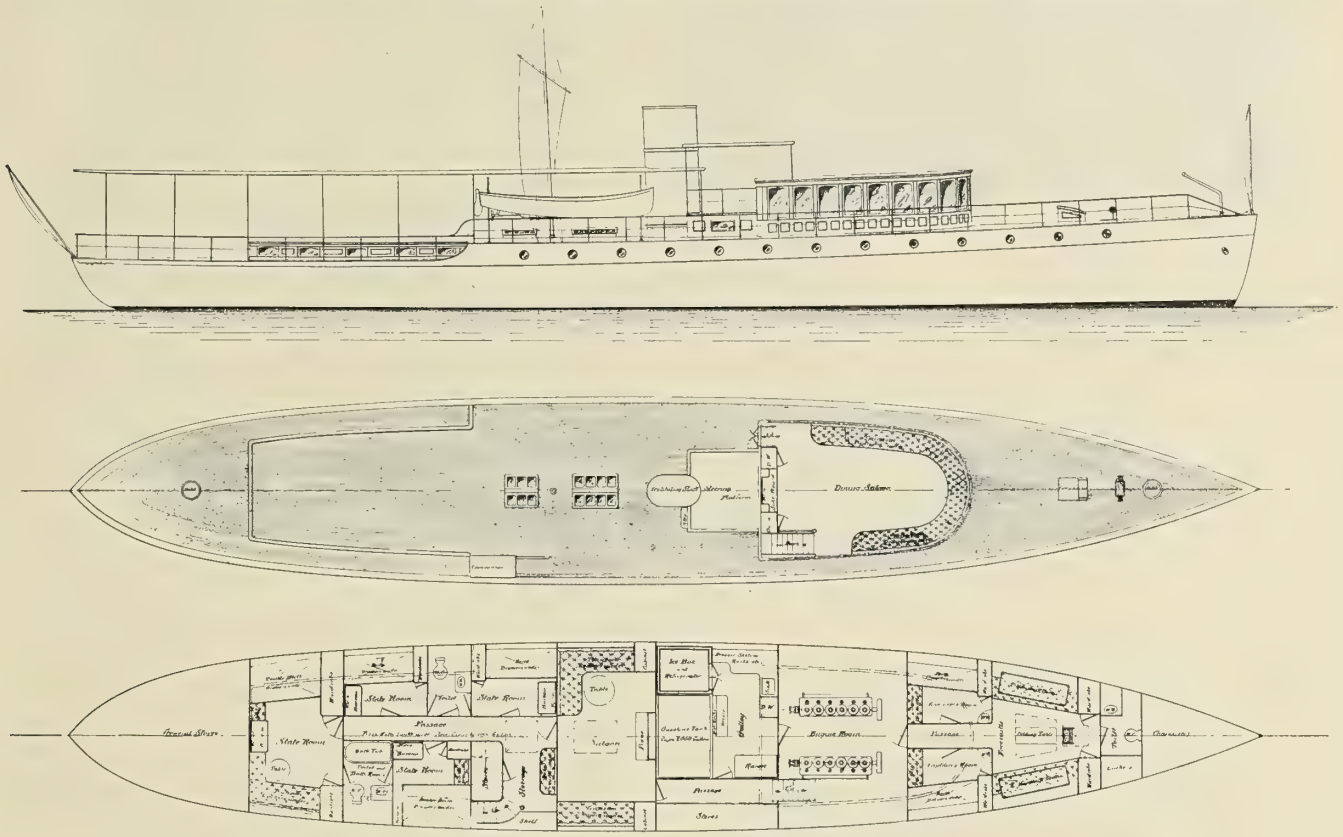
THE LIBRARY ON THE CRISTINA IS COSILY AND TASTEFULLY ARRANGED.—VIEW LOOKING TOWARDS PORT SIDE.
(Photograph, N. L. Stebbins, Boston.)

A New Petroleum Engine.

A French engineer, M. Sabathé, has in hand the preparing of designs for a torpedo boat destroyer of 500 tons displacement. The design calls for three screws, the outboard screws being operated by two engines placed in parallel, and the center screw by an engine placed forward of the others. The armament consists of two rapid-fire guns of 100 millimeters (4 inches), six guns of 75 millimeters (3 inches), and two torpedo tubes. Each engine has eight cylinders; combustion is effected by compression and not by electric spark. When cruising at 19-knot speed it is proposed to use the center screw only. For speeds above 19 knots and under 25 knots the center screw will be stopped, and the two outboard engines used. For speeds above 25 knots all three engines will be brought into play.

The designs call for a total engine weight of 220 tons, and at a speed of 14 knots the cruising radius is estimated at 4,800 nautical miles. The fore-and-aft length occupied by one of these engines of eight cylinders is 11 meters (36 feet), and the height from bed plate to top of cylinders is 2.9 meters (9½ feet). M. Sabathé estimates that 22 tons of petroleum will suffice for this vessel for a 1,200-mile run, assuming a 14-knot speed, and that the same vessel would use 52 tons of coal for this distance at the same speed. The calculations call for a petroleum consumption at this speed of 0.185 kilogram per horsepower per hour. It is estimated that this vessel will develop 850 horsepower effective when making 14 knots, and 9,200 horsepower effective when attaining a speed of 30 knots. An eight-cylinder engine of this design, when making about 300 turns per minute, will develop an effective horsepower of between 2,500 and 3,000.

For a vessel of 1,000 tons this engine can afford a maintained speed of 19 knots on an hourly consumption of petroleum of 0.19 kilogram per effective horsepower. For a reciprocating steam engine the coal consumption is estimated at 1.2 kilograms per effective horsepower. The designer says that these petroleum engines can be reversed in about twelve seconds without need of any special reversing engine. The cost of one of these 3,000-horsepower engines delivered at Havre is about 960,000 francs (£38,000, or $185,000).



OUTBOARD PROFILE, MAIN DECK AND LOWER-DECK PLANS OF TWIN SCREW STEEL MOTOR YACHT CRISTINA.

Shipbuilding.

Lloyd's report for the quarter ended September 30 shows under construction in the United Kingdom 319 merchant vessels, aggregating 733,378 gross tons. This is a loss of 66,000 tons, or 8 percent, from the June quarter, and is the lowest on record since the September quarter of 1896. It is but slightly more than half the figure for the September quarter of 1901 or the June quarter of 1906. A similar slump is shown in other countries, the construction under way in France being given as 44,803 tons; in Germany, 160,804 tons; in Italy, 43,070 tons; in Japan, 97,580 tons; and in the United States, 55,105 tons. This latter does not include the Great Lakes.

Under construction in the United Kingdom were shown 66 warships, aggregating 251,138 tons displacement. Eight of these are British and two Brazilian battleships; two, British armored cruisers; two British and two Brazilian scouts; while the balance are destroyers, submarines, etc.

The Bureau of Navigation in Washington reports the construction during the quarter ended September 30 of 319 vessels, aggregating 20,887 gross tons. Only thirteen of these vessels, aggregating 7,908 tons, were steel steamers. This is probably a low record for any quarter in the history of modern American shipbuilding. The corresponding figures for the 1907 quarter were 330 vessels of 133,092 gross tons, of which 36 vessels, aggregating 109,985 tons, were steel steamers. An interesting side light on the character of the construction this year is afforded by the fact that the steel steamers average only 608 tons, whereas last year the average was 3,055 tons, or more than five times as large.

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Rudders and Rudder Posts.

Some months ago we published in considerable detail a number of rudder designs from Pacific Coast practice of the United States and the practice of the Navy Department. These were largely warships, and all were large vessels.

In the present number we are fortunate enough to be able to supplement the previous article by another, covering practice on the Clyde in steamers of small and moderate size, including both screw, paddle-wheel and turbine propulsion.

The value of such articles, giving exact details on subjects of vital importance to the ship, can scarcely be over-estimated. The interchange of ideas resulting from the application to the problem of the minds of many engineers at widely separated points, and working under differing conditions, is of extreme value to everyone interested in the subject, and as such we commend both articles to careful perusal.

Society of Naval Architects and Marine Engineers.

The sixteenth general meeting of the Society will be held on Thursday and Friday, the 19th and 20th of November, in the Engineering Societies building, New York. A list of the seventeen papers prepared for discussion follows:

1. "The War Eagle," by Charles H. Cramp.
2. "Practical Methods of Conducting Trials of Vessels," by Col. E. A. Stevens.
3. "The Influence of Free Water Ballast upon the Stability of Ships and Floating Docks," by Naval Constructor T. G. Roberts, U. S. N.
4. "Further Experiments upon Longitudinal Displacement and its Effect upon Resistance," by Professor H. C. Sadler.
5. "Further Analysis of Propeller Experiments," by Clinton H. Crane.
6. "Deviation of the Compass Aboard Steel Ships; Its Avoidance and Correction," by Lieut.-Commander L. H. Chandler, U. S. N.
7. "The Influence of Midship Section Shape upon the Resistance of Ships," by Naval Constructor D. W. Taylor, U. S. N.
8. "Recent Inventions as Applied to Modern Steamships," by W. C. Wallace.
9. "Service Test on the Steamship *Harvard*," by Professor C. H. Peabody.
10. "Trials of the United States Scout Cruiser *Chester*, Fitted with Parsons Turbines," by Charles P. Wetherbee.
11. "Some Remarks on the Steam Turbine," by J. W. Powell.
12. "Shipbuilding on the Great Lakes," by Robert Curr.
13. "The Steamer *Commonwealth*," by J. H. Gardner and W. T. Berry.
14. "Fire Boats," by Charles C. West.
15. "Sea-Going Suction Dredges," by Thomas M. Cornbrooks.
16. "The British International Trophy Race of 1908," by W. P. Stephens.
17. "Transportation of Submarines," by Naval Constructor W. J. Baxter, U. S. N.

The papers are of unusual interest, three of them dealing with the newest and most prominent agent of ship propulsion—the steam turbine. In two of these three papers the subject of tests and trial trips is gone into extensively, and these two papers refer respectively to mercantile and war vessels. Two other papers, following in some measure papers by the same authors last year, relate to the resistance of ships; and both are based upon experiments with models conducted respectively in the government tank at Washington and the tank of the University of Michigan. The other papers are so scattered in general scope as not to be amenable to ready classification, but each is treated by an acknowledged authority on the subject, and the result ought to be a splendid list.

The Era of the Large Ship.

The advent of the Cunarders *Lusitania* and *Mauretania,* each of which has a gross tonnage more than 6,000 tons greater than any other ship afloat, was rightly heralded as marking a new era in the development of the large ocean steamship. Prior to these two ships the successive advances in tonnage had been relatively small, but continuous. Such a big jump as these two ships represented, however, was looked upon as setting the mark for some years to come. But their pre-eminence in this respect, it seems, is now challenged by the determination of the White Star Company to build two ships exceeding the two Cunarders in every dimension. These two vessels, which have already been named *Olympic* and *Titanic,* will be laid down by Harland & Wolff, in Belfast, next January.

Various statements have been made regarding their size; but it seems to be settled that they will be 860 feet long over all, and will have unusual beam and depth. The result is a ship with an estimated gross registered tonnage of 42,000, as compared with 30,822 for the *Lusitania* and 31,938 for the *Mauretania.* When we consider that, aside from the four ships mentioned, there has never been built a ship reaching as much as 25,000 gross tons, the tremendous stride in advance in this respect is at once apparent. The new ships, it is reported, will have a speed of about 21 knots, and will be furnished and decorated in most superior style. They are expected to be ready for the summer season of 1910, and will, naturally, be fitted to handle an enormous cargo.

It is interesting in this connection to note that the average length of the twenty largest steamships of the world in 1848 was only 230 feet; in 1873, twenty-five years later, the average was 390 feet; after another quarter-century the average length of the twenty largest was 541 feet, while it was 640 feet in 1903; in 1909, including the two new ships, the figure is 691 feet. The mean gross tonnage of these twenty largest ships is 24,285, as compared with 17,151 in 1903. If we average the lengths of the twenty *longest* ships, we have 700 feet; but three ships here included, being long and narrow, and intended largely for speed, do not appear among the twenty *largest,* reckoned in gross register tonnage.

	Ship.	Launched.	Line.	Length, Feet.	Gross Tonnage.
1	*Olympic**	1910	White Star	860	42,000
2	*Titanic**	1910	White Star	860	42,000
3	*Mauretania*	1906	Cunard	760	31,938
4	*Lusitania*	1906	Cunard	760	30,822
5	*George Washington†*	1909	North German Lloyd	700	24,800
6	*Kais. Augusta Victoria*	1905	Hamburg-American	678	24,581
7	*Adriatic*	1906	White Star	709	24,541
8	*Rotterdam*	1907	Holland-America	650	24,176
9	*Baltic*	1904	White Star	709	23,876
10	*Amerika*	1905	Hamburg-American	669	22,225
11	*Cedric*	1902	White Star	681	21,035
12	*Celtic*	1901	White Star	681	20,904
13	*Minnesota*	1904	Great Northern	622	20,718
14	*Caronia*	1905	Cunard	650	19,687
15	*Carmania*	1905	Cunard	650	19,524
16	*Kaiser Wilhelm II*	1902	North German Lloyd	684	19,361
17	*Kronprinzessin Cecilie*	1907	North German Lloyd	684	19,360
18	*President Grant*	1907	Hamburg-American	600	18,074
19	*President Lincoln*	1907	Hamburg-American	600	18,074
20	*Lapland†*	1908	Red Star	620	18,000
21	*Oceanic*	1899	White Star	686	17,274
22	*Prinz Friedrich Wilhelm*	1907	North German Lloyd	590	17,082
23	*Nieuw Amsterdam*	1906	Holland-America	600	16,913
24	*Deutschland*	1900	Hamburg-American	661	16,502
25	*Cincinnati†*	1908	Hamburg-American	580	16,400
26	*Cleveland†*	1908	Hamburg-American	580	16,400
27	*Arabic*	1903	White Star	601	15,860
28	*Republic*	1903	White Star	570	15,378
29	*Kronprinz Wilhelm*	1901	North German Lloyd	637	14,908
30	*La Provence*	1906	French	603	14,744

Ships not yet afloat are marked (*); those afloat but not yet in service (†). It will be noted that six of the first eleven are White Star ships, as are nine of the thirty. Seven belong to the Hamburg-American Line.

As a matter of interest, the thirty largest ships, averaging 665 feet in length and 21,575 tons gross register, are listed below. It may be noted that the *Cedric,* now eleventh on the list, was, as recently as 1903, the largest ship afloat. In 1899 this distinction was held by the *Oceanic,* now twenty-first in tonnage. A large steamer not listed is the *Europa,* of the Hamburg-American Line. This vessel has been projected for some time, but, we believe, is not actually under construction.

Theoretical Considerations.

The practical shipbuilder is prone to look with more or less scorn upon theoretical methods and processes. His work lies in transmuting the ideas of the designer, as represented by the drawings of a ship and her machinery and equipment, into a steel structure capable of battling with the elements of nature, and of carrying her load—be it passengers, cargo or warlike material—from place to place at a predetermined speed and under certain conditions of economy.

In the development of the design, however, it is necessary to investigate theoretically certain features regarding the adaptability of the proposed structure to the purposes intended. One of the most important of these has to do with the stability of the ship, and anything which would tend to decrease the onerous task of determining this directly from the lines and disposition of weights is welcomed as a boon by those who have the task in hand. An article on this subject in another column is quite opportune, and may be recommended to the careful consideration of naval architects generally.

Another matter, which is much less likely to receive attention than is the question of stability, is that concerning the adequacy of the riveted connections in the shell plating to take up their share in the burden imposed upon the structure of the ship by the motion of the waves, and by the relation between the buoyancy of the water, directed upwards, and the various component weights of the ship and her contents, directed downwards. An article on this subject will also be found in our pages this month, in which the matter is taken up theoretically but simply, and the subject is gone into in sufficient detail to show the methods employed in the Navy Department in Washington. It is obvious that the material and its connections should be properly proportioned to bear the loads to which they are to be subjected.

Progress of Naval Vessels.

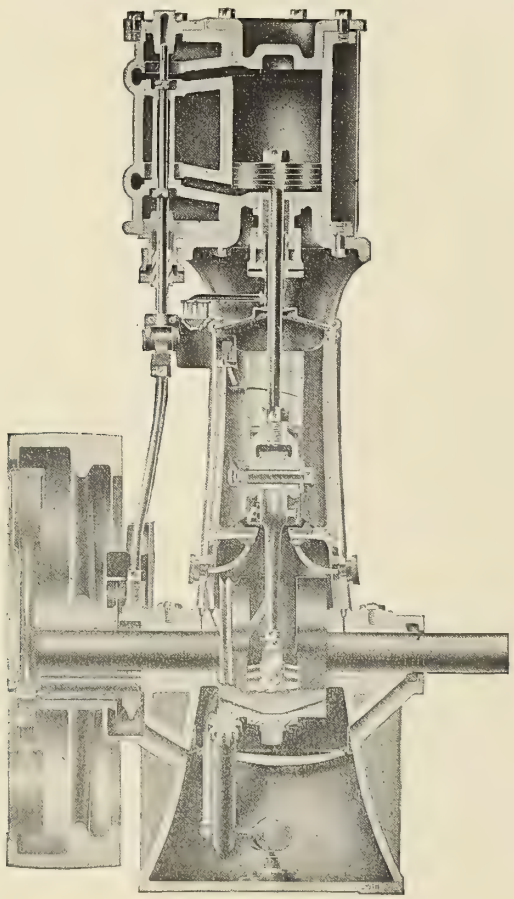
The Bureau of Construction and Repair, Navy Department, reports the following percentages of completion of vessels for the United States navy:

	Knots.	Tons.		Sept. 1.	Oct. 1.
BATTLESHIPS.					
S. Carolina..	16,000	18½	Wm. Cramp & Sons.....	58.1	63.2
Michigan ...	16,000	18½	New York Shipbuilding Co..	65.1	71.4
Delaware ...	20,000	21	Newp't News S.B.& D.D. Co.	40.5	44.2
North Dakota	20,000	21	Fore River Shipbuilding Co.	50.1	54.2
TORPEDO BOAT DESTROYERS.					
Smith	700	28	Wm. Cramp & Sons.....	49.3	53.8
Lamson	700	28	Wm. Cramp & Sons.....	46.7	52.1
Preston	700	28	New York Shipbuilding Co..	47.9	49.1
Flusser	700	28	Bath Iron Works.....	20.1	26.1
Reid	700	28	Bath Iron Works.....	20.1	26.0
SUBMARINE TORPEDO BOATS.					
Stingray	—	—	Fore River Shipbuilding Co..	55.7	58.7
Tarpon	—	—	Fore River Shipbuilding Co..	54.5	57.3
Bonita	—	—	Fore River Shipbuilding Co..	54.1	55.5
Snapper	—	—	Fore River Shipbuilding Co..	53.8	55.7
Norwhal	—	—	Fore River Shipbuilding Co..	48.2	50.9
Grayling	—	—	Fore River Shipbuilding Co..	47.9	50.7
Salmon	—	—	Fore River Shipbuilding Co..	46.8	50.3

ENGINEERING SPECIALTIES.

Economy Tests of High-Speed Engines.

In a paper read by C. H. Treat, before the American Society of Mechanical Engineers, he gave complete data on a test made July 9 and 10 on a 7 by 7-inch vertical self-oiling engine, which had been run almost continuously, night and day, for 10,000 hours, connected to a forced draft fan. The engine cylinder had been badly scored, and the mean diameter increased to 7.082 inches. The rings had been broken, due to water, and were replaced prior to test. The test shows results as low as 41 pounds of steam per indicated horsepower per hour under full load. This performance, it will be noted, is as good as many new engines in their best condition would show. A further observation leads to the belief that it is quite important from an economical point of view to keep the piston rings on small high-speed engines in shape to prevent undue leakage.



The initial tests on this engine were made at the plant of the American Blower Company, Detroit, Mich., in 1905. Test was made by condensing the exhaust in a surface condenser, consisting of steam heater section pipe immersed in a tank, having an overflow. The whole was up overhead, so that any drip or leaks could be seen and stopped. That the section itself did not leak at any time was shown by the entire absence of drip, unless steam was entering.

The condensation being piped down, heater pipes and base were slanted to drain through a water seal to prevent steam blowing into the upper of two barrels having drain valves at bottom, the lower barrel being on platform scales. For each barrel of water the weight before filling was subtracted from the full weight. The scales used weighed correctly. The indicator spring was calibrated by both the makers and observers. A long lever reducing motion was used to prevent lost motion. The drum spring was drawn tight, and at the high speeds cards were short. The engine was indicated at speeds up to 500 revolutions per minute; using a Prony brake at higher speeds and taking the brake horsepower. Readings were taken every five minutes after conditions had become constant and the engine was well warmed up. No corrections were made for moisture in steam, but a few determinations showed a quality of 98 to 99 percent.

The section of engine shows the arrangement of cylinder and valve chest, also the unique patented oiling system. Mr. Treat's test of the used engine gave the indicated friction horsepower as 1.18, this being only 4 percent of the indicated horsepower of the engine. The oiling system must receive much credit for keeping the internal friction at such a low percentage.

The Vixen Patent Milling File.

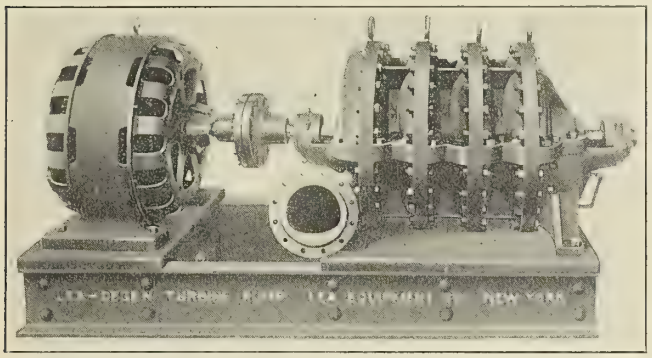
The "Vixen" patent milling file has semi-circular teeth on both sides, cut especially deep, which give to the filings the nature and shape of turnings or shavings produced by a modern lathe or milling machine. A special back, or holder, is used in conjunction with the file, which enables the mechanic to get a much greater purchase, and so to utilize to the fullest extent the superior cutting qualities of the file. The shape of the teeth not only makes the file act essentially as a milling cutter, but it also tends to keep the file clean, an object which is hard to accomplish with an ordinary file, especially when

working on soft metals. The shape of the teeth also prevents the file from slipping or chasing, and leaves a smooth, even surface. It is claimed that the file cuts equally well soft and tool steel, cast and wrought iron, bronze and all other hard metals, as well as brass, lead, aluminum and other soft metals. The file can be resharpened at least four times, each operation costing about half that of recutting an ordinary file. Moreover, it is claimed that, after resharpening, the file is quite as good as new, and, therefore, may be expected to last several times as long as an ordinary file. This file is manufactured by the National File & Tool Company, the Bourse, Philadelphia, Pa., and 8 White street, Moorfield, London, E. C.

A High-Duty Turbine Pump.

A pump which has been developed by the Lea Equipment Company, 136 Liberty street, New York, and is intended to be used largely in marine work, is operated by an induction or other type of electric motor, and runs through four stages, each of which is a separate unit and may readily be removed. It is said that no trouble is experienced from end thrust, the pump being so designed as to take care of this automatically. The present line of pumps includes those with suction and discharge from 3 to 24 inches in diameter.

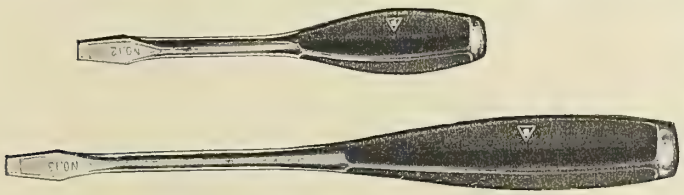
A 10-inch two-stage pump of this type, with eight blades, driven by a General Electric direct-current multipolar motor,



operated at 220 volts and with a capacity of 385 amperes, has been tested for efficiency at 400, 500 and 600 revolutions per minute. This pump was designed to deliver 3,000 gallons per minute, with a total head of 100 feet, at 600 revolutions per minute. The maximum efficiency at this speed was 77.97 percent when delivering 3,235 gallons per minute. The efficiency was above 70 percent for delivering all the way from 1,800 to 3,730 gallons per minute. At 500 revolutions the maximum efficiency of 77.62 percent was reached with a delivery of 2,794 gallons per minute. The efficiency was over 70 percent for all deliveries from 1,510 to 3,400 gallons per minute. At 400 revolutions per minute a maximum efficiency of 77.7 percent was reached with a delivery of 2,296 gallons per minute. The efficiency was above 70 percent for all deliveries from 1,250 to 2,850 gallons per minute.

An All-Steel Screwdriver.

A new and improved screwdriver, complete in one piece, is drop-forged of steel throughout, and the point is carefully tempered. The handle is of special design, insuring a positive and easy grip. There is nothing about it to loosen or get out



of order. It is simple, light, effective and durable. These screwdrivers are manufactured by the Billings & Spencer Company, Hartford, Conn., and are listed in eleven sizes, including two of heavier model, with square shank, for the application of a wrench.

PERSONAL.

Charles Ackerman, formerly general manager of the Crescent Shipyard (Lewis Nixon's) at Elizabethport, N. J., and later general manager of the Vacuum Cleaner Company, New York, is now general sales agent of the Mosher Watertube Boiler Company, with offices at 30 Church street, New York.

TECHNICAL PUBLICATIONS.

Nautical Charts. By G. R. Putnam, M. S. Size, 5¾ by 9 inches. Pages, 162. Figures, 49. 1908, New York: John Wiley & Sons. Price, $2.00; and London: Chapman & Hall, Ltd. Price, 8/6 net.

This work is based primarily on a lecture on charts, delivered before Columbia University. It deals with the methods of collecting and preparing information for the making of charts, their publication in the form of copper plates, their subsequent correction, and the reading and use of charts as an indispensable aid to navigation. It is shown in detail how soundings are taken and plotted, and how, from the original sounding charts, the finished charts are prepared, with various shadings for various depths of water, and selected soundings printed from the great mass taken. The various instruments used for this purpose, both for the actual soundings and the location of the point at which each sounding is taken, are illustrated and described, with considerable information regarding the process of triangulation, and the plotting of the results on various styles of projection, such as the mercator, the polyconic, the gnomonic and the arbitrary projection usually used for polar regions.

The subject of copper-plate engraving and printing is given quite a little attention, illustrations showing the methods of engraving the chart originally, and of engraving by means of a machine the various soundings on the copper plate. Electrotyping processes and lithographing processes are also illustrated, and etching on copper by means of a finished tracing and a sensitized copper plate is given some attention. In connection with the continual changes in channels, it is pointed out that the average loaded draft of the twenty largest steamships in the world increased from 19 feet in 1848 to 24 feet in 1873, 29 feet in 1898 and 32 feet in 1903. The average length of these vessels increased, respectively, from 230 feet to 390 feet, 541 feet and 640 feet.

Steam Power Plant Engineering. By G. F. Gebhardt, Professor of Mechanical Engineering, Armour Institute of Technology, Chicago. Size, 5¾ by 9 inches. Pages, 816. Figures, 461. 1908, New York: John Wiley & Sons. Price, $6.00 net; and London: Chapman & Hall, Ltd. Price, 25/6 net.

This book is the outcome of a series of lectures delivered to the senior class of the Armour Institute, and is primarily a text-book for engineering students, but also of interest to practicing engineers. The field embraced by the title is such a large one that it has been necessary to limit the treatment to essential elements only.

The work is divided into twenty-one chapters and six appendices, covering in rotation elementary considerations, fuels, boilers, furnaces and stokers, superheated steam, coal and ash-handling apparatus, chimneys, mechanical draft, steam engines, steam turbines, condensers, feed-water heaters, pumps, superheaters, piping, lubrication, finance and economics, instruments, typical specifications, a typical steam turbine station and a typical isolated station.

The work is full of references to original sources of information, particularly to current technical magazines; and the illustrations have been drawn from a great variety of sources, covering practice of all sorts all over the United States. Many of the illustrations are charts of performance, relating results to consumption of fuel and general cost of production and power.

In some respects the two last chapters, dealing, respectively, with the Commonwealth Edison Company, of Chicago, and the West Albany Power Station of the New York Central Railroad, are the most interesting of the work. Each deals with a thoroughly successful and up-to-date plant, the two plants considered being of totally different character, and markedly different in size. The installations are described in

some detail, while the illustrations give a good idea as to the layout and general equipment of the plants.

Ex-Meridian Altitude, Azimuth and Star-Finding Tables. By Lieutenant-Commander Armistead Rust, United States Navy. Size, 6¼ by 9½ inches. Pages, 393 + li. Figures, 13, including six folding plates. 1908, New York: John Wiley & Sons. Price, $5.00; and London: Chapman & Hall, Ltd. Price, 21/- net.

This work is not a text-book, and, therefore, rules for the conversion of time, the finding of hour angles and for plotting lines of position by the usual methods familiar to navigators, and which may be found in any work on navigation, have been omitted. A few examples have been worked, however, to illustrate the use of the various tables and diagrams.

The tables are so arranged that the "reduction to the meridian" may be found with sufficient accuracy for practical navigation at one opening of the book, while means are provided for readily obtaining the azimuth and correcting the reduction for variations in all of the elements when extreme accuracy is required. A new formula for the equation of altitudes for finding the chronometer error will, it is said, be found more convenient than that in general use, and the formula given for finding the longitude from ex-meridian observations has, it is believed, been reduced to such a simple form as to make it most useful when the weather has been too cloudy to obtain the longitude in any other way, or when the ship is in a region of strong currents.

The entire list of numbered pages is occupied by tables. In addition to these, twenty-five pages of the prefix are covered by tables of logarithms and corrections to be applied to the main tables. The first twenty-six pages of the prefix give methods for using the tables and plates and explanations of the same.

Verbal Notes and Sketches for Marine Engineers. By J. W. Sothern. Size, 5½ by 8½ inches. Pages, 438. Figures, 300. Glasgow, 1908: James Munro & Company. Price, 7/6 net and $2.25.

This is the sixth edition, and the main difference from the fifth is in the shape of additions covering indicator diagrams, valve settings, condensers, centrifugal pumps, steering gear, refrigeration, etc.

The work is in eight sections, covering respectively boilers, engines, general notes on materials and implements, indicator diagrams, marine electric lighting, refrigerating machinery, screw propellers and oil motors. It covers the subject from the point of view of the operating marine engineer, and is especially arranged to suit Board of Trade regulations, and to serve as a general reference book for marine engineers. It includes notes and sketches of verbal questions given at Board of Trade examinations to engineers competing for first-class and second-class certificates of competency.

The subject appears to be thoroughly practical in every sense, much of the material having been furnished by specialists along the particular lines represented. It includes a great variety of subjects, covering all sorts of material in which the marine engineer is interested, from the taking of indicator cards to the measurement of the pitch of a propeller, the adjustment of a lighting set and the operation of air compressors and ice machines. A short appendix gives the usual properties of saturated steam at pressures up to 200 pounds absolute per square inch.

The steamboat *New York*, of the Albany Day Line, one of the finest river steamers in the world, was totally destroyed by fire early in the morning, October 21. The vessel was built in 1887, was 335 feet long, and propelled by paddle wheels at a maximum speed of about 21 miles per hour.

QUERIES AND ANSWERS.

Questions concerning marine engineering will be answered by the Editor in this column. Each communication must bear the name and address of the writer.

Q. 418.—Is there any convenient formula for estimating the weights of marine boilers from known data of the boilers? Y. K. N.

A.—A paper read in 1897 before the Society of Naval Architects and Marine Engineers, and republished in our columns in February and June of 1898, gives formulas for Scotch, low cylindrical and locomotive boilers, including their respective mountings. Separate formulas are given for water in these boilers. The formula in each case covers one boiler only.

For single-ended Scotch boilers, the weight in tons (2,240 pounds) is $0.0075\ [D^2\ (L + 2) + H.\ S.] + 1.5$.

For double-ended Scotch boilers, weight $= 0.0075\ [D^2\ (L + 2) + H.\ S.]$.

For locomotive boilers, weight $= 0.0075\ [D^2\ (L + 2) + H.\ S.] - 1.5$.

For single-ended low cylindrical boilers, weight $= 0.005\ [D^2\ (L + 2) + H.\ S.] + 8$.

For double-ended low cylindrical boilers, weight $= 0.005\ [D^2\ (L + 2) + H.\ S.] + 6.5$.

For water in the main boilers other formulas are used. In Scotch boilers, with water 8 inches above the top row of tubes, the weight $= 0.0117\ (D^2\ L - A)$.

In low cylindrical boilers, with water 6 inches above top row of tubes, weight $= 0.01\ (D^2\ L - B)$.

In locomotive boilers, with water 6 inches above the furnace crown, weight $= K\ (D^2\ L - C - 125)$.

For auxiliary boilers, including contained water, the weight, for both horizontal and vertical types, may be placed at $0.031\ D^2\ L$.

In the above formulas the following notation is used:

D = Mean diameter in feet.
L = Length over end plates in feet.
$H.\ S.$ = Total heating surface of the boiler in square feet.
$T.\ S.$ = Tube heating surface in square feet.

$$A = \frac{H.\ S. \times d}{12\ \pi} + nf^2\ l.$$

$$B = \frac{H.\ S. \times d}{12\ \pi} + nf^2\ (L - l).$$

$$C = \frac{T.\ S. \times d}{12\ \pi}$$

d = Diameter of tubes in inches.
l = Length of tubes in feet.
f = Maximum diameter of furnaces in feet.
n = Number of furnaces in one boiler.
K = 0.028 for boilers with dry bottom and one furnace; 0.03 for two furnaces; 0.032 for boilers with wet bottom and one furnace, and 0.034 for two furnaces.

It will be noted that the tons per inch at above levels of water will be approximately $0.0021\ D\ L$.

These formulas are part of a large number, going into the estimated weights of machinery for both warships and merchant steamers. They have been computed from the recorded detail weights of machinery of more than three hundred vessels, embracing almost all of the types found in service, and have been found to give very close results in practice. In a number of cases, where Scotch, locomotive and low cylindrical

boilers were fitted, the estimated weights of machinery complete were found to come within 3½ percent of the actual recorded weights. In computations for 115 separate ships there were only eleven cases where the error exceeded the above percentage.

SELECTED MARINE PATENTS.

The publication in this column of a patent specification does not necessarily imply editorial commendation.

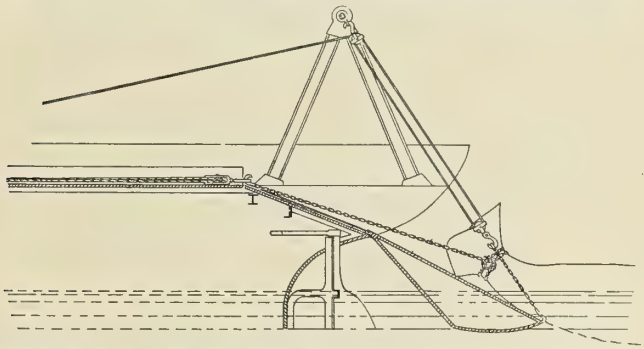
American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Loan & Trust Building, Washington, D. C.

893,324. SAFETY DEVICE FOR VESSELS. PAUL JAMNITZKY, CONEMAUGH, PA.

Claim 1.—The combination of a vessel having compartments formed therein, of air compressors, reservoirs for receiving air from said compressors, pipes for supplying air to the numerous compartments of said vessel, frames arranged longitudinally upon the bottom of said vessel, members at the ends of said frames for providing a compartment beneath said vessel when aground, hoods arranged in the bottom of said vessel and communicating with said reservoir, gates arranged in said hoods for controlling the direction of the discharge of air in said hoods, and a frame arranged upon the deck of said vessel for discharging air and dispelling fog surrounding said vessel. Three claims.

893,642. VESSEL FOR WHALE FISHING. JENS A. MORCH, CHRISTIANIA, NORWAY.

Claim 3.—In combination, a hull provided with a deck and having its stern recessed to form a receiving portion inclined downwardly from



said deck, an extension proportioned to close and fill said receiving portion when in a non-operating position, and adapted to form a continuation of said receiving portion when in an operating position, means mounted on said hull for operating said extension, and means mounted on said hull for drawing a load over said extension and inclined portion onto said deck. Five claims.

896,037. LIFE-SAVING APPARATUS. JOHN W. NEELY, CAMBRIDGE, MASS., ASSIGNOR OF ONE-HALF TO JOHN H. SMITH, JAMAICA PLAIN, MASS.

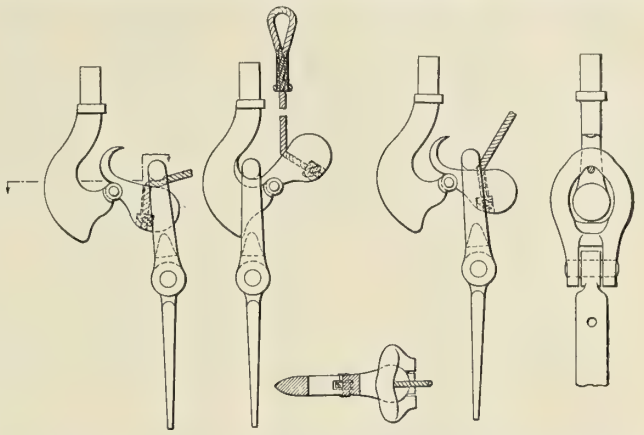
Abstract.—The invention relates to improvements in life-saving apparatus, and the object is to provide a device which may be worn by a person who is either boating or who is near the water, said device comprising a float and a cord fast to said float coiled and supported on a belt in such a manner that if the person's body becomes submerged the float will free itself from the belt and rise to the surface, thus indicating the position of the body, so that the body may be raised by means of the cord. Four claims.

896,349. STOCKLESS ANCHOR. WALTER S. BICKLEY, CHESTER, PA.

Abstract.—The invention has for its object to improve this class of anchor and to strengthen the head. A further object of the invention is to produce an anchor of this character that can readily be made of cast steel, which will make it equal to a wrought iron anchor, being ductile and of equal strength of forging. One claim.

894,345. DEVICE FOR ATTACHING AND DETACHING BOATS. JAMES R. RAYMOND, BAYONNE, N. J.

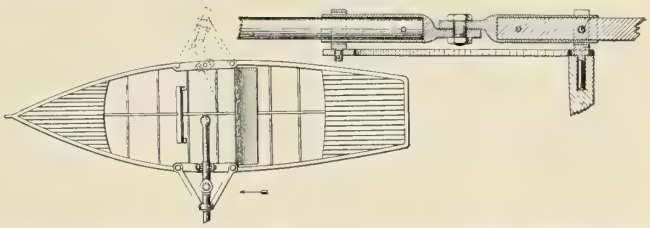
Claim 1.—The combination in a boat attaching and detaching device, of a movable hook having a neck and an enlarged bill, a shackle adapted



to engage said neck, and a lanyard adapted to guide said shackle to place on said neck and secured to said hook at a substantial distance back of said enlarged bill, whereby the lanyard in hooking on prevents direct engagement between said bill and shackle, and guides said shackle on to the neck of said hook. Six claims.

895,863. ROWING APPLIANCE. WILLIAM F. JAMES AND WILLIAM T. DEACON, VELASCO, TEXAS.

Claim 1.—In a rowing appliance for row-boats, a bracket consisting of outwardly-converging arms connected together and having means connecting the free ends thereof to the boat, permitting of a limited swinging movement of the bracket vertically, an oar-lock supported in the



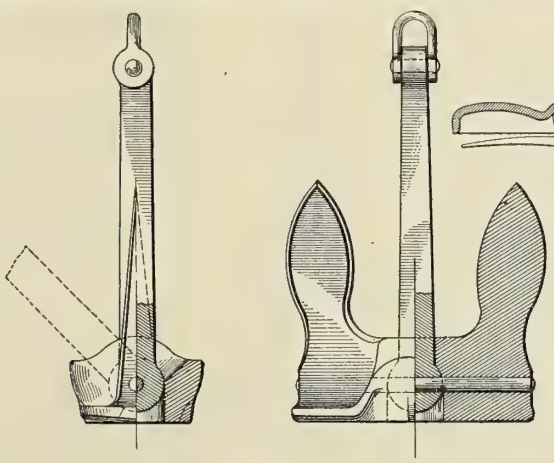
connecting portions of said arms, an oar-lock revolubly supported by the boat between said arms, and an oar passing through the oar-locks, composed of two sections pivoted together between said locks, adapting the sections of the oar to swing laterally. Four claims.

898,094. BOAT-LASHING DEVICE. FRED T. CLAYTON, SANDWICH, MASS., ASSIGNOR OF ONE-HALF TO ROBERT A. HAMMOND, SANDWICH, MASS.

Claim 1.—A device for lashing a boat to the deck of a vessel, comprising in its construction a link, the opposite ends of which are adapted to be pivotally connected, respectively, to a fixture fast to said boat and a fixture fast to the deck of said vessel, said link consisting of a rod, a latch pivoted to one end of said rod, said latch having a cam-shaped inner edge adapted to bear against said first-named fixture for the purpose specified, and means to lock said latch to said rod. Two claims.

897,341. PROPELLER-ADJUSTING DEVICE. WILLIAM E. BLAIR, BUFFALO, N. Y., ASSIGNOR TO BUFFALO GASOLINE MOTOR CO.

Claim 1.—A propeller-adjusting device comprising a main driving shaft, propeller blades adjustably supported on said shaft, a shifting rod movable lengthwise of the shaft and operatively connected with the propeller blades, an adjusting shaft arranged at an angle to the driving shaft and adjusting rod and operatively connected with said rod, a counter shaft arranged parallel with the adjusting shaft, a large gear secured to the adjusting shaft, a small gear secured to the counter shaft and meshing with the large gear, and a hand lever connected with the counter shaft. Five claims.



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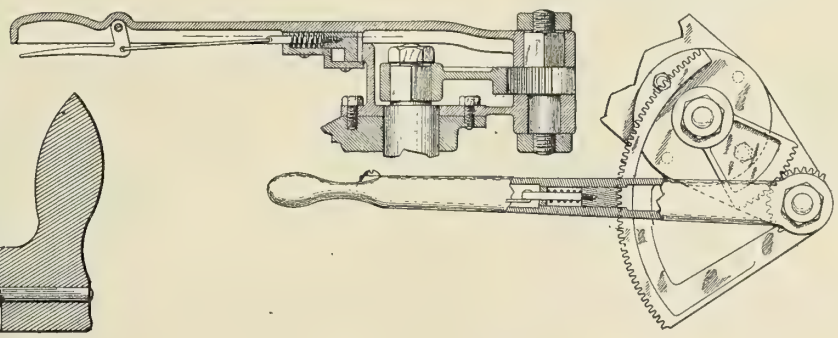


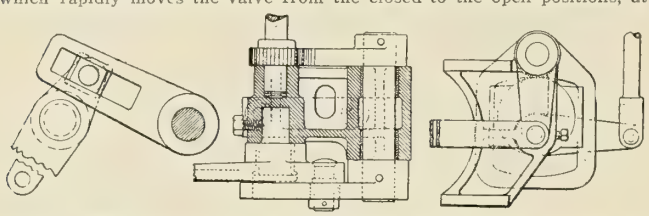
897,341.

British patents compiled by Edwards & Co., chartered patent agents and engineers, Chancery Lane Station Chambers, London, W. C.

4,576. ELASTIC-FLUID TURBINES. C. H. RAWSON, AND BUCHHOLTZ REVERSIBLE TURBINE SYNDICATE, LONDON.
In a multiple expansion reversible turbine, rotatable disks mounted on a shaft alternate with fixed disks secured to the casing. An annular series of blade passages is formed in each rotatable disk, and similar guide passages are formed in the fixed disks. To allow for expansion, the radial depth of the passages increases in a step-like manner, or the increase may be gradual. The turbine is made reversible by means of a ring of perforations in each rotatable disk, and a corresponding ring of perforations in each fixed disk, the former being inclined to the plane of the disk, and the latter being parallel to it. Oppositely arranged inlets and outlets are provided for forward and reversed driving respectively.

4,839. STEERING GEAR. T. L. LIVINGSTON, JARROW-ON-TYNE.
Two driving means are provided for operating the rudder, both of which are always in gear and may be brought into operation either separately or simultaneously. The modifications comprise means whereby the stop nut on one motor is actuated by operating the other motor, in such a manner that the traveling stop nuts in connection with each motor always correspond with each other and the angle to which the rudder is moved.

5,264. STEERING ENGINES. W. G. GIBBONS, EDINBURGH.
The controlling gear transmits its motion to the valve by mechanism which rapidly moves the valve from the closed to the open positions, at

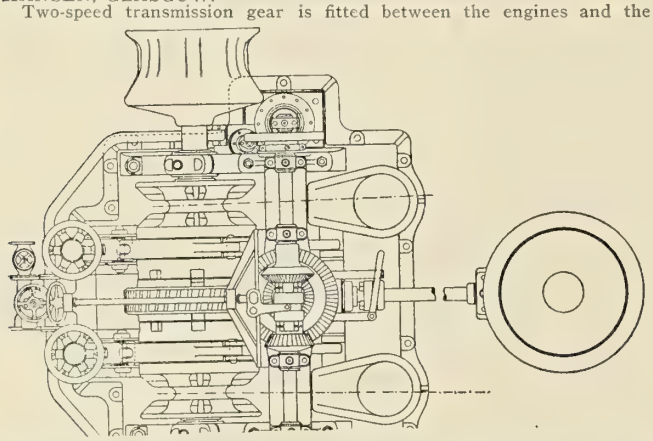


which positions the gear is free to continue its movement without substantially moving the valve. The controlling-valve lever is slotted, and is operated by a block on the pin attached to the small arm of the crank. In the open position the movement of the crank has substantially no effect on the valve. In a modification, the spindle operates a second spindle through a toothed sector and pinion. The second spindle is connected to the controlling valve. In another arrangement the movement of the lever is transmitted to a bridle by an eccentric. The bridle is connected to the valve by a spindle, lever and links.

6,321. RAISING SUNKEN SHIPS. B. REINIER, VIDAUBAN, FRANCE.
A cylinder for raising sunken ships is sunk by opening an air cock. Water then enters through a siphon tube, which is bent so as to form a water seal in any position of the cylinder; or water may be allowed to enter through a cock. When the cylinder is full, the cock is closed. In order to refloat the cylinder, air is forced in through a second siphon tube, water escaping through the first tube. Water in either end of the cylinder runs into the manhole chamber through tubes, and thence to the outside through the exit. Cocks are provided for screwing on connecting pipes for the combined working of several cylinders.

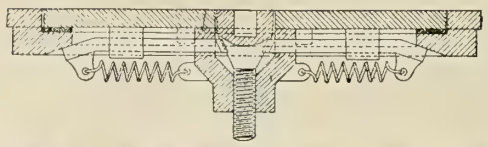
6,640. TURBINES. BELLISS & MORCOM, AND R. K. MORCOM, BIRMINGHAM.
Relates to making and fixing blades for turbine rotors and stators. Thin metal stampings are bent so as to form the front and back plates or surfaces of two adjacent blades connected by a foot, the plates being also united by one or more thin distance pieces; the back plate may have a strengthening rib. The blades are formed by assembling a number of these blade elements, the feet being secured in undercut grooves by bifurcated retaining blocks and a stringer, which spreads the fangs and flattens the arched foot. The widths of the front and back plates may vary, so that the edge of the blade, which may be sharpened, may have one or two thicknesses of metal.

6,889. WINDLASSES; CAPSTANS. A. KELLY AND C. D. B. HANSEN, GLASGOW.
Two-speed transmission gear is fitted between the engines and the



horizontal drum shaft of a windlass or capstan. The horizontal shaft carrying the cable drums and warping drums is driven from the engines through two-speed bevel and worm gearing. The worm is mounted on a vertical shaft carrying toothed wheels at its upper end, engaging the loosely-mounted bevel wheels on the engine shaft.

6,999. SHIPS' HATCHES AND HATCHWAYS. P. B. CLARKE, LIVERPOOL.
Relates to ships' hatches, etc., in which the covers are flush with the

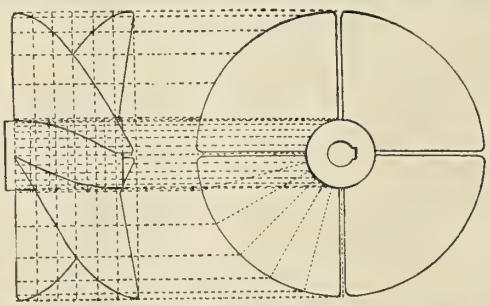


deck when in place, and the framing is provided with an inclined surface which is adapted to be engaged by the inclined outer ends of the sliding locking-bolts. To the deck is fixed a ledge, and between this and the hatch cover is a strip of india-rubber packing. The bolts are adapted to slide in guides in the hatch cover and the fitting. When the tapered bolt is screwed down, it pushes the other bolts outwards and forces their downwardly inclined edges under and against the inclined faces of the ledges, thus making a tight joint.

7,647. SHIPS. A. F. YARROW, POPLAR, LONDON.
In order to keep ships' cabins cool in hot climates, the tops are covered with loose canvas, woven reeds, or other suitable material that absorbs moisture. One or more revolving sprayers supply water from the river or sea to the material, and the evaporation serves to keep the surrounding space cool.

7,955. TURBINES. A. J. B. LEGE, ISLINGTON, LONDON.
Impact wheels are arranged for multiple expansion. A piston part or bucket attached to a disk is acted on by steam introduced through nozzles. After acting on the bucket the steam passes through nozzles in the fixed ring and impinges against similar buckets or piston parts. It next passes through nozzles to other buckets, and exhausts through ports. The disk carries piston parts on both sides. The piston parts may be formed with nozzle-like apertures so that fluid passing through reduces the pressure in front of the piston part. Labyrinth packing is provided where necessary. For reverse driving, steam is admitted to a central chamber, and is delivered by nozzles to buckets. It exhausts through ports, or may pass through openings in the ring to the general exhaust, or may even act further on the ahead buckets.

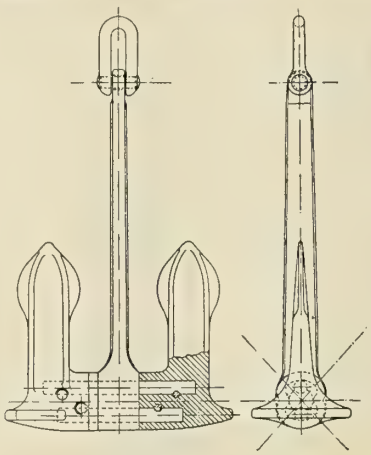
8,049. SCREW PROPELLER. D. McLACHLAN, CARDIFF.
The blades of the propeller are helical and have radial leading edges



and inclined trailing edges. The leading and trailing edges are parallel to one another or nearly so, when viewed end on, and spring from a cylindrical boss.

9,201. SCREW PROPELLER. J. STRAKA, TRIESTE, AUSTRIA.
In the type of propeller in which the blades are surrounded by and secured to a circumferential ring, the blades are constructed with sharp and nearly straight leading edges and curved following edges, the width of the blades decreasing to the center of the propeller.

9,211. ANCHOR. J. B. RICHARDSON AND D. S. HOLLOWAY, PONTYPRIDD, GLAMORGAN.
The head is divided into two parts, united by one or more bolts, which pass through a recess in the shank. The bolts are secured to the arms by pins, or they may pass right through the arms and be riveted or otherwise secured at the ends. To avoid bending of the bolts a block is fitted in the shank, and is of such shape that the side of the chamber always contacts with it. The bolts may be in one with the block.



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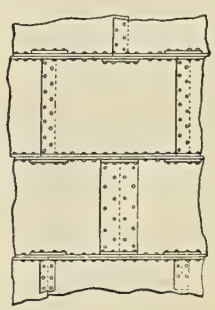

9,280.

9,280. CHANNEL-PLATE CONSTRUCTIONS. A. JEFFREY AND J. V. KEANE, COTTESLOO, WESTERN AUSTRALIA.
Ships' hulls, bulkheads and decks, and other analogous plate structures are built up of plates of channel section, riveted or bolted together through the flanges, which thereby form ribs to the structure. The ends of the plates are riveted together by lap joints, or by butt joints reinforced by strap plates. In the construction of ships' hulls, the joints may run either longitudinally or vertically.